APPLIED HEAT TRANSMISSION

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XI

PREFACE

This book is intended for both practicing engineers and students. Its purpose is to put into readily usable form some of the more important data on heat transmission and to describe some of the common types of heat-transfer equipment and kinds of insulation used in industry. As indicated by the title, emphasis is placed on practical applications rather than on theory.

The first three chapters deal with the equations required for the solution of elementary problems involving heat transfer by conduction, radiation, and convection. It is hoped that the method of presentation used is one that will be particularly attractive to practicing engineers, who are likely to be more concerned with methods of solution than with mathematical derivations. Thus, in each section the equations are stated first, the methods of applying them are then illustrated by the solution of numerical problems, and finally the derivation of the equations is explained.

Attention is directed particularly to the charts and tables presented in Chaps. IV and V. By means of these charts and tables, convection coefficients and values of the pressure drop for a number of fluids under various conditions can be quickly and easily determined. Although the accuracy of some of the equations upon which these charts and tables are based is not yet definitely established, it is believed that in every case the values given are conservative and may be safely used for design purposes. The method of applying these charts and tables is illustrated by the solution of numerical problems at the end of each chapter.

The last two chapters are devoted to a description of some of the more common types of heat-transfer equipment and kinds of insulation used in industry. Equations for calculating the thickness of insulation required for several special conditions of industrial importance are given at the end of the last chapter.

The application of these equations is also illustrated by the solution of several numerical problems

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INTRODUCTION

Heat may be transferred by conduction, by convection, and by radiation. These three methods may be defined as follows:

Conduction is the transfer of heat from one particle of matter to another, the particles remaining in fixed positions relative to each other.

Convection is the transfer of heat from one part of a fluid to another by the mixing of the warmer particles of the fluid with the cooler. For example, consider a pan of water placed on a hot stove. Heat is transferred through the pan and to the particles of water in contact with the pan by conduction. These particles, becoming more buoyant as they are heated, rise and mix with the main body of the water, thus transferring heat by convection. Such motion of the fluid, caused entirely by differences in density within the fluid, is called *natural convection*. If the motion of the fluid is produced by some mechanical means, such as a stirrer, pump, or fan, it is called *forced convection*.

Radiation is the transfer of heat from one body to another as the result of the bodies' emitting and absorbing a form of energy called *radiant energy*. All matter possesses the property of both emitting and absorbing such energy to a greater or lesser degree, the effect on the matter being the same as though heat were absorbed from or added to it.[1] Radiant energy travels through space at the velocity of light, light being radiant energy of those wave lengths to which the human optic nerve is sensitive. Unlike heat, radiant energy does not require the presence of any matter for its transmission. Thus, the earth receives radiant energy from the sun even though separated from it by an almost perfect vacuum.

[1] Except in the rare cases where photochemical reactions are involved

In practice, heat transfer by more than just one of the three methods is usually involved For example, consider the heat transfer from an ordinary steam "radiator" to the air in a room The flow of heat through the metal walls of the radiator and to the air in contact with the surface of the radiator takes place by conduction As the air surrounding the radiator is heated, it rises and carries heat to the rest of the room by convection At the same time the radiator emits radiant energy, most of which is transmitted through the air to the walls of the room The walls absorb part of this radiant energy and convert it into heat and reflect the rest back to the radiator or to other parts of the wall The total rate at which the radiator loses heat is equal to the rate at which it loses heat by convection plus the rate at which it loses heat by radiation, and this total rate is necessarily equal to the rate at which heat flows through the walls of the radiator by conduction

Since all three methods of heat transfer are likely to be involved, it is necessary to be able to calculate the rate of heat transfer by each of the three methods in order to design heat-transfer equipment It frequently happens, however, that even though all three methods may be involved, the rate of heat transfer by one of the methods is small and may be neglected For example, the effect of radiation may usually be neglected in designing shell-and-tube type equipment

CONDUCTION

1. Thermal Conductivity.—The calculation of the rate at which heat flows through any material by conduction involves a property of the material called the *thermal conductivity*. This property is defined as the *rate* at which heat flows through the material by conduction per unit of cross-sectional area taken normal to the direction of heat flow and per unit temperature gradient measured in the direction of heat flow. Consider, for example, a small element of the material having a cross-sectional
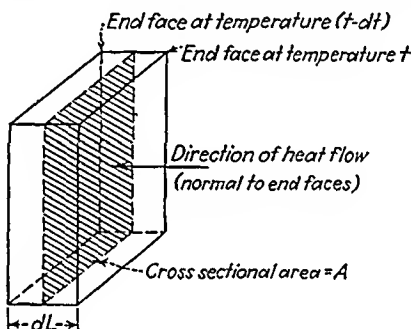
Fig. 1.—Unidirectional conduction of heat.

area A and thickness dL, as shown in Fig. 1. If the two end faces of the element are maintained at the uniform temperatures t and $(t - dt)$, and if the sides of the element are surrounded by other material at the same temperature so that no heat is lost laterally, heat will flow at some rate q from the face at the higher temperature to the face at the lower temperature. Since the end faces are both at uniform temperatures, the direction of heat flow will always be normal to these faces. By definition, the thermal conductivity k of the material is equal to the rate of heat flow q divided by A (the cross-sectional area normal to the direction of heat flow) and divided by $-dt/dL$ (the temperature gradient in the direction of heat flow); *i.e.*,

$$k = \frac{q}{A(-dt/dL)} \qquad (1)$$

The thermal conductivities of various materials differ widely and can be determined only experimentally   Tables of thermal conductivities of various materials are given in the Appendix

The units of k depend upon the units chosen for each of the quantities in the right-hand member of Eq (1)   Thus, if the rate of heat flow q is measured in B t u per hour, if the cross sectional area A is measured in square feet, and if the temperature gradient dt/dL is measured in degrees Fahrenheit per foot, the net units of k are

$$\frac{\dfrac{\text{B t u}}{\text{hr}}}{\text{ft}^2 \times \dfrac{°\text{f}}{\text{ft}}} = \frac{\text{B t u}}{(\text{ft})(\text{hr})(°\text{F})}$$

These are the units that are used throughout the book   The thermal conductivities of many engineering materials, particularly industrial insulating materials, are frequently expressed in the units B t u /(hr)(sq   ft)(°F /in)   However these units are inconvenient in calculating the rate of heat transfer by conduction through bodies of any shape other than flat   Values of k expressed in such units can be converted to the units B t u /(ft)(hr)(°F) by dividing them by 12
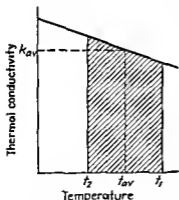
Fig 2 —Variation of thermal conductivity with temperature

2   Effect of Temperature and Pressure on Thermal Conductivity —The thermal conductivities of practically all materials depend upon the temperature of the material   For some materials the thermal conductivity increases as the temperature of the material rises   for others it decreases   This variation is approximately linear over a considerable range of temperature for most materials   Thus, if the thermal conductivity is plotted against temperature as in Fig 2 a straight line is obtained

For practical purposes the thermal conductivities of all materials are unaffected by changes in pressure   At extremely

high pressures the thermal conductivities of liquids do increase, and at very low absolute pressures the thermal conductivities of gases decrease; but these pressures are outside the range usually encountered in engineering work.

3. Steady and Unsteady Conduction of Heat.—The conduction of heat through any body is said to be *unsteady* if the temperature at any point in the body varies with time, and it is said to be *steady* if the temperature at every point in the body remains constant. Hence, with unsteady conduction the temperature gradient dt/dL at each point in the body also varies with time; with steady conduction the temperature gradient at each point remains constant. For example, when steam is first admitted to a pipe covered with insulation, the temperature at the inner surface of the insulation rises very rapidly, but the temperature at any point near the outer surface is not immediately affected. The temperature at each point in the insulation continues to rise for some time, and during this period the flow of heat through the insulation is unsteady. Finally, however, the temperature at each point reaches some value at which it remains constant, and the flow of heat is then steady.

4. Fundamental Equation for Conduction.—Equation (1) can be written

$$q = -kA \frac{dt}{dL}. \tag{2}$$

This equation, known as *Fourier's law,* is the fundamental equation for unidirectional heat transfer by conduction. The equation applies to fluids as well as to solids, provided that no convection currents exist within the fluids. It is impossible to obtain this condition, however, except in very thin layers of the fluid, and consequently the rate of heat transfer through fluids must ordinarily be calculated by the equations for convection given in Chap. III.

Equation (2) must be integrated before it can be used to solve practical problems. If the conduction of heat is unsteady, this integration is usually quite difficult, because the temperature gradient dt/dL at each point in the body varies with time, and therefore the rate of heat flow q also varies with time. Fortunately, many practical problems involve only steady conduction. For steady conduction, the rate of heat flow q is constant because the temperature gradient dt/dL at each point in the body

remains constant Consequently, for steady conduction the integration of Eq (2) requires only (1) that the area A of the isothermal surfaces existing within the body be expressed as a function of the distance L to these surfaces and (2) that the thermal conductivity k of the body be expressed as a function of the temperature t Even this integration is difficult, however, unless the area A is some simple function of the distance L

Integrated forms of Eq (2) for the steady conduction of heat through several types of bodies frequently encountered in practice are given in the next three sections These equations all involve the surface temperatures of the bodies In practice these temperatures are frequently not known, but instead only the temperatures of the media in contact with the surfaces of the body are known For example, in the case of insulation on piping the temperature of the air surrounding the pipe is always known, but the temperature of the outside surface of the insulation itself is usually not known Surface temperatures can be determined by making use of the fact that, for steady conduction the rate at which heat is transferred through the body by conduction is equal to the rate at which heat is transferred from the surfaces of the body to the surrounding media by radiation and convection This method of calculation is explained in detail in Sec 61 in Chap VII In the present chapter it is assumed that the surface temperatures of the bodies are known

An approximate method of calculating temperature gradients and values of q for the unsteady conduction of heat through flat bodies is explained in Sec 10

✓6 **Conduction of Heat through Flat Bodies** —If the conduction of heat is steady, the rate of heat transfer through homogeneous flat bodies can be calculated by the equation

$$q = \frac{kA(t_1 - t_2)}{L} \qquad (3)$$

where q = the rate of heat transfer by conduction, B t u per hr

k = the thermal conductivity of the material at the aver age of temperatures t_1 and t_2 B t u /(ft)(hr)(°F)

A = the cross-sectional area of the body, taken normal to the direction of heat flow, sq ft

t_1 and t_2 = the temperatures at the two faces of the body, °F

L = the thickness of the body, ft

Illustrative Problem 1.—Calculate the rate of heat transfer by conduction through each square foot of a wall 8 in. thick made of calcined, diatomaceous silica insulating bricks if the temperatures at the two faces of the wall are 1200 and 200°F.

Solution.—The average of the temperatures at the two faces of the wall is 700°F., and the thermal conductivity of calcined, diatomaceous silica bricks at this temperature is 0.15 B.t.u./(ft.)(hr.)(°F.)   Hence

$$q = \frac{0.15 \times 1 \times (1200 - 200)}{\frac{8}{12}}$$
$$= 225 \text{ B.t.u. per hr.}$$

Equation (3) can be obtained from Eq. (2) as follows: Since steady conduction is assumed, q is constant; and since the body is flat, A is constant.   Hence, Eq. (2) can be written

$$\frac{q}{A} \int_0^L dL = - \int_{t_1}^{t_2} k \, dt.$$

The right-hand member is represented by the shaded area shown in Fig. 2.   If k is assumed to vary linearly with temperature, this area is equal to $k_{av}(t_1 - t_2)$, where k_{av} is the value of k at the arithmetic average of t_1 and t_2.   Therefore, integrating,

$$\frac{q}{A} L = k_{av}(t_1 - t_2),$$

which is Eq. (3).

6. Conduction of Heat through Cylindrical Bodies.—If the conduction of heat is steady, the rate of heat transfer through homogeneous bodies whose surfaces are *concentric cylinders* can be calculated by the equation

$$q = \frac{k 2 \pi l (t_1 - t_2)}{2.3 \log_{10} (r_2/r_1)}, \qquad (4)$$

where q = the rate of heat transfer by conduction from the inner to the outer surface, B.t.u. per hr.

k = the thermal conductivity of the material at the average of temperatures t_1 and t_2, B.t.u./(ft.)(hr.)(°F.).

π = 3.14.

l = the length of the cylinder, ft.

t_1 and t_2 = the temperatures at the inside and outside faces respectively, °F.

r_1 and r_2 = the radii (or the diameters) of the inside and outside faces respectively, ft. or in.

If t_2 is greater than t_1, a negative value is obtained for q, which indicates that the direction of heat flow is from the outer to the inner surface

It is interesting to note that since only the ratio of the radii appears in Eq (4), the rate of heat transfer by conduction per foot of length is independent of the absolute size of the cylinder Thus, if the surface temperatures of the insulation are the same, the rate of heat loss per foot of length from a 1-in diameter tube covered with ½ in of insulation is the same as the rate of heat loss from a 6 in diameter tube covered with 3 in of the same insulation

It is frequently possible to use Eq (3) in place of Eq (4) without introducing any serious error Thus if the ratio of r_2/r_1 does not exceed 2, substituting an arithmetic average of the inside and outside areas for A in Eq (3) will give a result within 4 per cent of that given by Eq (4)

Illustrative Problem 2—Calculate the rate of heat loss per foot of length from a 3-in pipe covered with 1 in of 85% Magnesia if the temperature of the inside surface of the insulation is 500°F and the temperature of the outside surface is 100°F

Exact Solution—The average of the temperatures of the inside and outside surfaces of the insulation is 300°F and the thermal conductivity of 85% Magnesia at this temperature is 0 043 Btu /(ft)(hr)(°F)   The actual outside diameter of a 3-in pipe is 3 500 in and therefore the outside diameter of the insulation is 5 500 in   Hence by Eq (4)

$$q = \frac{0\ 043 \times 2 \times 3\ 14 \times 1 \times (500 - 100)}{2\ 3 \log_{s} (5\ 500 \ 3\ 500)}$$

$$= 240 \text{ Btu per hr}$$

Approximate Solution—Since the ratio of r_2/r_1 is less than 2 the rate of heat loss can be calculated approximately by Eq (3)

$$q = \frac{0\ 043 \times 3\ 14 \times (4\ 500/12) \times 1 \times (500 - 100)}{\tfrac{1}{12}}$$

$$= 243 \text{ Btu per hr}$$

Equation (4) can be obtained from Eq (2) as follows The area A of any cylindrical element taken as shown in Fig 3 is equal to $2\pi L L$ Since steady conduction is assumed q is constant and Eq (2) can be written

$$\frac{q}{2\pi l} \int_{r_1}^{r_2} \frac{dL}{L} = - \int_{t}^{t_2} k \, dt$$

As in Sec. 5, if k varies linearly with the temperature, the right-hand member is equal to $k_{av}(t_1 - t_2)$. Hence, integrating,

$$\frac{q}{2\pi l} \log_e \left(\frac{r_2}{r_1} \right) = k_{av}(t_1 - t_2),$$

which is Eq. (4).

7. Conduction of Heat through Spherical Bodies.—If the conduction of heat is steady, the rate of heat transfer through
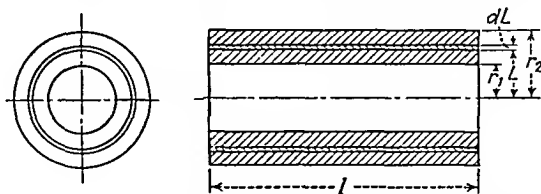

FIG. 3.—Cross section of a homogeneous, hollow cylindrical body.

homogeneous bodies whose surfaces are *concentric spheres* can be calculated by the equation

$$q = \frac{k4\pi r_1 r_2 (t_1 - t_2)}{r_2 - r_1}, \tag{5}$$

where q = the rate of heat transfer by conduction from the inner to the outer surface, B.t.u. per hr.

k = the thermal conductivity of the material at the average of temperatures t_1 and t_2, B.t.u./(ft.)(hr.)(°F.).

π = 3.14.

t_1 and t_2 = the temperatures at the inside and outside faces respectively, °F.

r_1 and r_2 = the radii of the inside and outside faces respectively, ft.

As with Eq. (4), a negative value of q indicates that the direction of heat flow is from the outer to the inner surface.

It may be noted that, whereas the rate of steady conduction through either flat or cylindrical bodies is zero if the bodies are infinitely thick, the rate of steady conduction through spherical bodies never becomes less than a certain limiting value, regardless of how thick the walls may be. Thus, if r_2 is infinite, Eq. (5) reduces to

$$q = k4\pi r_1 (t_1 - t_2) \tag{5a}$$

Illustrative Problem 3—Liquid oxygen is frequently stored in spherical containers because the ratio of outside surface to volume is a minimum for a sphere (1) Calculate the rate at which heat leaks into such a container 5 ft in diameter insulated with a 1 ft thick layer of powdered diatomaceous silica packed to a density of approximately 10 lb per cu ft, if the inner surface of the insulation is at −290°F and the outer surface is at 50°F (2) What minimum rate of heat leak could be obtained by increasing the thickness of the insulation?

Solution—1 Since the thermal conductivities of all porous insulating materials decrease as the temperature of the insulation is lowered, the thermal conductivity of powdered diatomaceous silica packed to the given density, and at an average temperature of −120°F is less than the value 0 026 B t u /(ft)(hr)(°F) at 70°F Assuming a value of 0 022 B t u / (ft)(hr)(°F),

$$q = \frac{0.022 \times 4 \times 3.14 \times 2.5 \times 3.5 \times (-290 - 50)}{1}$$

$$= -820 \text{ B t u per hr },$$

the negative sign indicating that the direction of heat flow is into the container

2 If the insulation were made infinitely thick, the rate of heat leak would be, by Eq (5a),

$$q = 0.022 \times 4 \times 3.14 \times 2.5 \times (-290 - 50)$$

$$= -235 \text{ B t u per hr}$$

Equation (5) can be obtained from Eq (2) in a manner exactly analogous to that used in deriving Eq (4)

8. Conduction of Heat through Bodies of Any Shape

—The integration of Eq (2) for homogeneous bodies of any shape other than those discussed in the three preceding sections is quite difficult, and consequently it is usually impossible to calculate the exact rate of heat transfer by conduction through such bodies A satisfactory approximation can frequently be obtained, however, by using an arithmetic average of the inside and outside areas for A in Eq (3) This procedure gives too high a value for the rate of heat transfer, although the error is small if the thickness of the body (and consequently the difference between the inside and outside areas) is relatively small The importance of this error is frequently minimized by the presence of other inaccuracies Thus, the thermal conductivity of the material may not be accurately known, the body may not be perfectly homogeneous, or the temperatures at the two faces may not be absolutely uniform

Illustrative Problem 4—A cylindrical tank 15 in in diameter and 5 ft long is insulated with 3 in of felted rock wool If the inside surface of the

insulation is at 640°F. and the outside surface is at 120°F , calculate the
rate at which heat is lost from the tank

Approximate Solution —The inside area of the insulation is 22 0 sq ft ;
the outside area is 35 0 sq ft ; and the average of these two areas is 28 5
sq ft   The average of the inside and outside surface temperatures is 380°F.,

Fɪɢ 4 —Composite wall of a brick kiln in process of construction   *(Courtesy
of the Johns-Manville Co)*

and the thermal conductivity of rock wool at this temperature is 0 045
B t u /(ft)(hr)(°F)   Hence, by Eq (3),

$$q = \frac{0\ 045 \times 28\ 5 \times (640 - 120)}{\frac{3}{12}}$$
$$= 2670 \text{ B t u per hr.}$$

For the special case of rectangular bodies having relatively
thick walls, as in small furnaces, an average area to be used in
Eq. (3) can be calculated by empirical equations given by
Langmuir.[1]

[1] *Trans Am Electrochem Soc* , vol 24, p 53, 1913

9 Conduction of Heat through Several Bodies in Series — Furnace walls, the insulation on high temperature piping, and many other types of equipment are frequently constructed of layers of several different kinds of materials through which heat flows in series by conduction   For example, Fig 4 shows a brick kiln wall consisting of an inner layer of firebrick, a core of insulating brick, and an outer layer of red brick   The firebrick is used to protect the insulating brick from mechanical abrasion
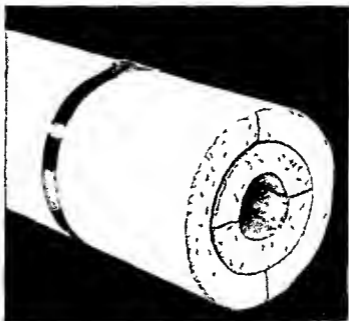

Fig 5 —Combination high temperature pipe insulation   (Courtesy of the Keasbey & Mattison Co)

and from the high temperatures existing within the kiln   The red brick is cheaper than the insulating brick but has a higher thermal conductivity and cannot be used at the high temperatures existing at the center of the wall   Figure 5 shows a type of insulation suitable for covering high-temperature piping   It consists of an inner layer of high temperature insulation and an outer layer of 85% Magnesia   The 85% Magnesia has a lower thermal conductivity than the high temperature insulation but cannot be used at temperatures above 600°F   The rate of

heat transfer by conduction through such composite bodies can be calculated as follows:

If the conduction of heat is steady, the rate of heat transfer through two *flat bodies* in series, as shown in Fig. 6, can be calcu-
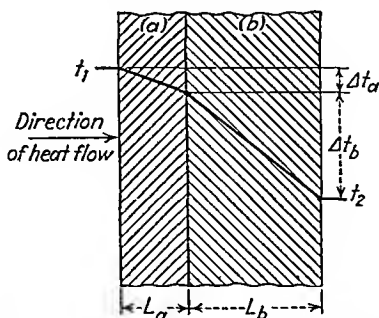

FIG. 6.—Temperature gradient through two flat bodies in series.

lated by the equation

$$q = \frac{A(t_1 - t_2)}{(L_a/k_a) + (L_b/k_b)}, \tag{6}$$

where q = the rate of heat transfer by conduction, B.t.u. per hr.

k_a and k_b = the thermal conductivities of materials a and b, evaluated at the average temperature of each, B.t.u./(ft.)(hr.)(°F.).

A = the cross-sectional area of the bodies, taken normal to the direction of heat flow, sq. ft.

t_1 and t_2 = the temperatures at the outside faces of the composite body, °F.

L_a and L_b = the thicknesses of materials a and b, ft.

This equation may also be written

$$q = \frac{(t_1 - t_2)}{R_a + R_b}, \tag{6a}$$

where R_a and R_b are equal to L_a/k_aA and L_b/k_bA, respectively, and are called the *resistances* of the two materials. These two equations can be extended to include any number of flat bodies in series by adding additional L/k terms to the denominator of Eq. (6) or resistances R to the denominator of Eq. (6a).

If the conduction of heat is steady, the rate of heat transfer through two *cylindrical bodies* in series, as shown in Fig. 7, can be calculated by the equation

$$q = \frac{2\pi l(t_1 - t_2)}{2\,3\left[\dfrac{\log_{10}\,(r_a''/r_a')}{k_a} + \dfrac{\log_{10}\,(r_b''/r_b')}{k_b}\right]} \qquad (7)$$

where q = the rate of heat transfer by conduction from the
         inner to the outer surface, B t u per hr

k_a and k_b = the thermal conductivity of materials a and
         b, evaluated at the average temperature of each,
         B t u /(ft)(hr)(°F).

       π = 3 14

       l = the length of the cylinders, ft

  t_1 and t_2 = the temperatures at the inside and outside faces
         of the composite body, °F

r_a' and r_a'' = the radii (or the diameters) of the inside and out-
         side faces of material a, ft or in

r_b' and r_b'' = the radii (or the diameters) of the inside and out-
         side faces of material b, ft or in

This equation can be extended to include any number of con
centric cylindrical bodies in series
by adding additional $(1/k)\log_{10}$
(r''/r') terms to the denominator

Fig 7 —Temperature gradient
through two cylindrical bodies in
series

In applying Eqs (6) or (7), the
thermal conductivity of each of
the materials should be evaluated
at the average temperature of the
material, and consequently the
temperatures between the various
layers must be determined
These temperatures may also be
required for design purposes if
some of the materials are suitable
for only a limited range of tem
perature They can be determined as follows (1) A reasonable
value for each temperature is first assumed (2) Based on these
assumed values, the thermal conductivity of each of the materials
is determined (3) The rate of heat transfer q through the com-
posite body is calculated by Eq (6) or (7) (4) Using this value of
q, the temperatures at each of the interfaces is calculated by Eq (3)
or (4) These calculated temperatures will usually be sufficiently
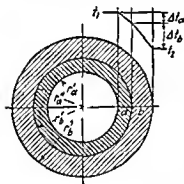accurate for practical purposes, although the procedure may be

repeated if greater accuracy is desired. The method of calculation is illustrated in the following problems:

Illustrative Problem 5.—A furnace wall consists of 9 in. of firebrick covered with 4 in. of insulating blocks made of diatomaceous silica, asbestos fiber, and a bonding material. Calculate the rate of heat transfer by conduction through each square foot of the wall if the temperature at the inside face is 2000°F. and the temperature at the outside face is 200°F.

Solution.—As a first approximation, assume that the temperature at the interface between the two materials is 1600°F. Based on this assumption, the average temperatures of the firebrick and of the insulation are 1800 and 900°F., respectively, and the thermal conductivities of the two materials at these temperatures are 0.71 and 0.061 B.t.u./(ft.)(hr.)(°F.). By Eq. (6), the rate of heat transfer through the composite wall is

$$q = \frac{1 \times (2000 - 200)}{\frac{9}{12 \times 0.71} + \frac{4}{12 \times 0.061}}$$
$$= 276 \text{ B.t.u. per hr.}$$

Since this is also the rate of heat transfer through each individual material, a more accurate value of the temperature t' at the interface can be calculated by Eq. (3). Thus, for the insulation,

$$276 = \frac{0.061 \times 1 \times (t' - 200)}{\frac{4}{12}},$$

whence

$$t' = 1710°F.$$

The same result would be obtained by applying Eq. (3) to the firebrick. This temperature does not agree very closely with the temperature originally assumed, but an appreciably different value of q would not be obtained by repeating the calculations, since the thermal conductivities of the two materials vary only slightly with the temperature. Thus, a value of 280 B.t.u. per hr. is obtained if the calculations are repeated.

Illustrative Problem 6.—A 2-in. pipe is to be covered with two layers of insulation each 1 in. thick. An insulating material made of diatomaceous silica, asbestos, and a bonding material is to be used for the inner layer; and 85% Magnesia, which is suitable only for temperatures up to 600°F., is to be used for the outer layer. Will this covering be satisfactory if the temperature at the inside face will be 1000°F. and the temperature at the outside face will be 120°F.?

Solution.—As a first approximation, assume that the temperature at the interface will be 600°F. Based on this assumption, the average temperatures of the inner and outer layers will be 800 and 360°F., and their thermal conductivities at these temperatures will be 0.059 and 0.044 B.t.u./(ft.)(hr.)(°F.), respectively. The actual outside diameter of a 2-in. pipe is 2.375 in. Hence, by Eq. (7), the rate of heat transfer per foot of length will be

$$q = \frac{2 \times 3 \, 14 \times 1 \times (1000 - 120)}{2 \, 3 \left[\dfrac{\log_{10}(4 \, 375/2 \, 375)}{0 \, 059} + \dfrac{\log_{10}(6 \, 375/4 \, 375)}{0 \, 044} \right]}$$

$$= 292 \text{ B t u per hr}$$

Since this is also the rate of heat transfer through the individual layers, the temperature t at the interface can be calculated by Eq (4) Thus, using the data for the outer layer

$$292 = \frac{0 \, 044 \times 2 \times 3 \, 14 \times 1 \times (t - 120)}{2 \, 3 \log_{10}(6 \, 375/4 \, 375)},$$

whence

$$t' = 520°F$$

Although a more accurate value could be obtained by repeating the calculations the foregoing value is sufficiently accurate to indicate that the 85% Magnesia will not be overheated and that the covering will therefore be satisfactory

Equations (6) and (7) involve the assumption that no drop in temperature takes place at the boundary between the two materials In practice, however, the contact between the layers is usually not perfect because of the roughness of the surfaces, and consequently a drop in temperature does take place As a result, the actual rate of heat transfer by conduction is likely to be somewhat less than the calculated rate

Equation (6) can be obtained as follows Referring to Fig 6, the rate of conduction through material a is, by Eq (3),

$$q = \frac{k_a A (\Delta t_a)}{L_a},$$

or

$$\Delta t_a = \frac{q L_a}{A k_a}$$

Similarly,

$$\Delta t_b = \frac{q L_b}{A k_b}$$

Since steady flow is assumed the rate of heat transfer q is the same through both materials Hence, adding the last two equations,

$$t_1 - t_2 = \frac{q}{A} \left(\frac{L_a}{k_a} + \frac{L_b}{k_b} \right),$$

which is Eq (6)

Equation (7) is obtained from Eq (4) in a similar manner An equation for calculating the rate of heat transfer by conduction

through two spherical bodies in series can be obtained from Eq. (5), but this case is of little practical importance.

10. Unsteady Conduction of Heat through Flat Bodies.—If the temperature at each surface of a homogeneous flat body remains uniform over the entire surface but changes with time, and if the temperatures within the body at some initial time are known, the temperatures within the body at any subsequent time can be
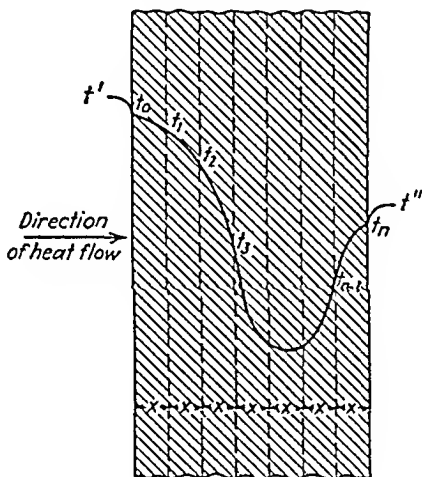

Fig. 8.—Temperature gradient for unsteady conduction through a homogeneous flat body.

closely approximated by the following method developed by E. Schmidt:[1]

1. The body is divided into n imaginary laminae of equal thicknesses x, as shown in Fig. 8. Any value may be chosen for n; the larger the value the greater the accuracy of the method.

2. The temperatures $t_0, t_1, t_2, \ldots, t_n$ at the boundaries of these n imaginary laminae are to be determined at successive, equal intervals of time θ, the value of θ being calculated by the equation

$$\theta = \frac{x^2}{2}\left(\frac{c\rho}{k}\right), \tag{8}$$

[1] SCHMIDT, E., "A. Föppls Festschrift," p. 179, J. Springer, Berlin, 1924.

where θ = the time interval, hr
      x = the thickness of each imaginary lamina, ft
      ρ = the density of the material, lb per cu ft
      c = the specific heat of the material, B t u /(lb)(°F)
      k = the thermal conductivity of the material (assumed constant), B t u /(ft)(hr)(°F)

3 A table is prepared with the headings shown in Table 1, the vertical columns being the temperatures t_0, t_1, t_2,   , t_n,

TABLE 1—CALCULATION FORM FOR UNSTEADY CONDUCTION THROUGH FLAT BODIES

Elapsed time hours	Temperatures -°F							t_{n-1}	t_n
	t_0	t_1	t_2	t_3	t_4				
0									
θ									
2θ									
3θ									
4θ									
5θ									

and the horizontal rows being the values of these temperatures after the time intervals θ, 2θ, 3θ, etc , have elapsed The known initial temperatures are entered in the first horizontal row

4 Each of the temperatures in the body of the table is equal to the arithmetic average of the two temperatures indicated by the arrows in Table 1

Case 1—If the surface temperatures t_0 and t_n at the successive intervals of time θ, 2θ, 3θ etc , are known, the vertical columns of t_0 and t_n can be filled in immediately and each of the temperatures in the body of the table can then be calculated

Case 2—If the surface temperatures t_0 and t_n of the body are not known but the temperatures t' and t'' of the media in contact with the two surfaces are kn n, the values of t_0 and t_n can be calculated by the equations

$$t_0 = \frac{h_t'xt' + kt_1}{k + h_t'x} \quad \text{and} \quad t_n = \frac{h_t''xt'' + kt_{n-1}}{k + h_t''x} \quad (9)$$

in which all the temperatures are evaluated for the same instant, and
t_0 and t_n = the surface temperatures of the body, °F.

t_1 and t_{n-1} = the temperatures within the body at distances x from the two surfaces, °F.

t' and t'' = the temperatures of the media in contact with the two surfaces, °F.

h_t' and h_t'' = the "combined coefficients" for convection and radiation from the surfaces of the body to the surrounding media (see Table 59 in Chap. VII), B.t.u./(sq. ft.)(hr.)(°F.).

x = the thickness of the imaginary laminae, ft.

k = the thermal conductivity of the material, B.t.u./(ft.)(hr.)(°F.).

After the temperatures existing at the boundaries of the imaginary laminae at successive intervals of time have been determined, the rates of heat transfer into or out of the body at any instant can be approximated by applying Eq. (3) to the outside laminae. The method of calculation is illustrated by the following problem:

Illustrative Problem 7.—An industrial furnace is to be operated 5 days a week but is to be shut off and allowed to cool down over the week end. It is to be electrically heated and is to operate at a temperature of 1800°F. Hence, the walls, which are to be 9 in. thick, may be constructed either of firebrick or of calcined, diatomaceous silica insulating brick. It is estimated that, in starting up, $2\frac{1}{2}$ hr. will be required to raise the temperature of the inside surface of the wall from its initial temperature of 80°F., the ambient air temperature, to its operating temperature of 1800°F. In order to determine which construction will be the more economical (*i.e.*, in order to determine for which material the fixed charges plus the operating charges will be the smaller), the total heat loss from each square foot of surface per 120-hr. week is to be calculated for each material.

Data.—Evaluating the thermal conductivities of the two materials at an assumed average wall temperature of 1000°F., for the firebrick $k = 0.67$ B.t.u./(ft.)(hr.)(°F.), $c = 0.29$ B.t.u./(lb.)(°F.), and $\rho = 125$ lb. per cu. ft.; and for the insulating brick $k = 0.16$ B.t.u./(ft.)(hr.)(°F.), $c = 0.23$ B.t.u./(lb.)(°F.), and $\rho = 38$ lb. per cu. ft. The combined coefficient h_t'' for convection and radiation from the outside surface of the wall to the surrounding air will increase as the temperature of the surface of the wall increases, but an average value of 3.0 B.t.u./(sq. ft.)(hr.)(°F.) may be assumed.

Solution.—If the wall is divided into $n =$ four imaginary laminae, $x = \frac{9}{12} \div 4 = 0.1875$ ft. Considering first the firebrick, by Eq. (8),

$$\theta = \frac{(0\ 1875)^2 \times 0\ 29 \times 125}{2 \times 0\ 67}$$

$$= 0\ 95\ \text{hr}$$

Assuming that the temperature t_0 of the inside surface of the wall rises at a uniform rate during the first 2½ hr of operation, the values of t_0 at the end of 0.95 and 1 90 hr can be determined by interpolation Hence, the top horizontal row and the first vertical row of temperatures in Table 2 can be

TABLE 2 —TEMPERATURE GRADIENTS AND RATES OF HEAT LOSS THROUGH
FIREBRICK WALL

Elapsed time, hr	Temperature, °F					Rate of heat loss q, B t u /(sq ft) (hr)
	t_0	t_1	t_2	t_3	t_4	
0	80	80	80	80	80	0
0 95	734	80	80	80	80	2340
1 90	1388	407	80	80	80	3510
2 85	1800	731	244	80	80	3810
3 80	1800	1022	407	162	80	2780
4 75	1800	1104	592	244	169	2490
5 70	1800	1196	674	381	243	2160
6 65	1800	1237	789	459	286	2010
7 60	1800	1295	848	538	328	1805
8 55	1800	1324	917	588	356	1700
9 50	1800	1359	956	637	382	1575
∞	1800				475	1185

filled in immediately All the temperatures for the first 4 75 hr with the exception of the last value of t_4 can then be calculated by finding the arithmetic average of the two preceding temperatures indicated in Table 1 The last value of t_4 can be calculated by Eq (9) Thus,

$$t_4 = \frac{3\ 0 \times 0\ 1875 \times 80 + 0\ 67 \times 244}{0\ 67 + 3\ 0 \times 0\ 1875}$$

$$= 169°F$$

All the temperatures in the next horizontal row, with the exception of t_4, can then be calculated by averaging the preceding temperatures, and t_4 can again be calculated by Eq (9) Proceeding in this way, all the temperatures in the rest of the table can be determined

The rate of heat loss q from the interior of the furnace can be calculated by applying Eq (3) to the inside lamina For example, the rate of heat loss from each square foot of surface after the elapse of 3 80 hr is

$$q = \frac{0\ 67 \times 1 \times (1800 - 1022)}{0\ 1875}$$

$$= 2780\ \text{B t u per hr}$$

After the wall reaches a state of equilibrium and steady conduction is established, the rate at which heat is transferred through the wall by conduction is equal to the rate at which heat is lost from the surface of the wall to the surrounding air. Thus, for each square foot of surface,

$$q = \frac{0.67 \times 1 \times (1800 - t_4)}{\frac{9}{12}} = 3.0 \times 1 \times (t_4 - 80).$$

Hence, $t_4 = 475°F.$, and $q = 1185$ B.t.u. per hr.

The values given in Table 3 for the insulating brick are calculated in exactly the same manner.

If the rates of heat loss q are plotted as ordinates against time as the abscissa, the area under the curve obtained is equal to the total heat loss.

TABLE 3.—TEMPERATURE GRADIENTS AND RATES OF HEAT LOSS THROUGH INSULATING BRICK WALL

Elapsed time, hr.	Temperature, °F.					Rate of heat loss q, B.t.u./(sq. ft.) (hr.)
	t_0	t_1	t_2	t_3	t_4	
0	80	80	80	80	80	0
0.96	747	80	80	80	80	570
1.92	1415	414	80	80	80	855
2.88	1800	748	247	80	80	899
3.84	1800	1024	414	164	80	663
4.80	1800	1107	594	247	117	592
5.76	1800	1197	677	356	141	515
6.72	1800	1239	777	409	152	479
7.68	1800	1289	824	465	164	436
8.64	1800	1312	877	494	171	417
9.60	1800	1339	903	524	177	393
∞	1800	....	...	...	193	342

This is found to be 154,700 B.t.u. per sq. ft. per week for the firebrick and 43,300 B.t.u. per sq. ft. per week for the insulating brick.

It is interesting to note that, with the exception of approximately the first half hour of operation, the rate of heat loss from the interior of the furnace is always greater than the rate eventually reached when steady conduction is established. It may also be noted that, whereas the heat loss through the firebrick wall for steady conduction can be reduced by adding insulation to the outside surface of the wall, this insulation would have no effect whatsoever on the heat loss from the interior of the furnace during the first 6.65 hr. of operation.

Other simplified methods of solving unsteady-flow problems have been developed.[1] The advantage of the method given here

[1] *Ind. Eng. Chem.*, vol. 15, p. 1173, 1923. See also MARGARET FISHENDEN and O. A. SAUNDERS, "Calculation of Heat Transmission," His Majesty's

is the fact that it is applicable regardless of the manner in which the surface temperatures of the body may vary or what the initial temperature gradient may be   It has the disadvantage of being somewhat laborious if either too large a value is chosen for n or the temperature gradient after a long period of elapsed time is desired

Stationery Office (London), 1932 or A SCHACK, ' Industrial Heat Transfer ' translated by H Goldschmidt and E P Partridge, John Wiley & Sons Inc , 1933

CHAPTER II

RADIATION

11. Introduction.—Radiant energy is emitted by all matter, the rate of emission depending upon the temperature and the nature of the matter. Unlike heat energy, radiant energy does not require the presence of any matter for its transmission. Thus, the earth receives radiant energy from the sun even though separated from it by an almost perfect vacuum. On encountering any matter, radiant energy may be partly absorbed by the matter and converted into heat energy; it may be partly reflected by the matter; or it may be transmitted through the matter. Consider, for example, a bare steam pipe running through a room. The radiation emitted by the pipe is almost all transmitted through the air in the room but is either absorbed or reflected by the walls. The walls, on the other hand, also emit radiation, part of which falls upon the pipe and is partly absorbed. The net rate at which the pipe loses heat by radiation is equal to the rate at which it emits radiation, minus the rate at which it absorbs radiation from the surrounding walls. Thus, the net rate at which any solid body gains or loses heat by radiation depends not only upon the temperature and the nature of its own surface but also upon the temperature and the nature of the surrounding surfaces.

Regarding the effect of the nature of the surfaces, it has been found experimentally that different surfaces *all at the same temperature* emit radiant energy at different rates but that there is a maximum rate that none of them exceeds. It has also been found that different surfaces absorb different fractions of the radiant energy incident upon them. The hypothetical body whose surface would absorb all the radiation incident upon it is called a *black body*. It can be shown that such a body also has the important property of emitting radiation at the maximum possible rate at each temperature.

The problem of calculating the net rate of heat transfer by radiation between two bodies whose surfaces are at uniform

temperatures and are separated by some medium that absorbs only a negligible portion of the radiant energy passing through it[1] is treated in this chapter. A method of solving such problems is explained in the next section, and the theory upon which this method is based is discussed in the remaining sections of the chapter.

12 Calculation of Heat Transfer by Radiation. *a Emissivity*—In order to calculate the net rate at which any body gains or loses heat by radiation, it is necessary to know the *emissivity* of its surface. This quantity may be defined either as the ratio of the rate at which the given surface emits radiation to the rate at which a black body at the same temperature would emit radiation or as the ratio of the rate at which the given surface absorbs the radiation incident upon it to the rate at which a black body would absorb the same radiation, these two ratios being equal.[2] Thus, the emissivity of a surface may be thought of as its "efficiency" so far as its ability to emit or absorb radiation is concerned. The emissivity of most surfaces is not constant but increases with increase in temperature. Values of the emissivity for a number of surfaces are listed in Table IV in the Appendix.

Fig 9.—
Radiation from
a grooved
surface

b Effective Area—The rate at which any body gains or loses heat by radiation depends also upon the shape of its surface. Consider for example, a body having a plane surface traversed by a groove, as shown in Fig 9. Assuming uniform temperature each element of the surface emits radiation at the same rate, but since part of the radiation from each element of the surface of the groove is intercepted and absorbed by the walls of the groove, less heat is lost from the body per unit area of the grooved surface than per unit area of the plane surface. Hence,

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[1] Most gases absorb very little radiant energy although water vapor carbon dioxide, sulfur dioxide methane and ammonia are important exceptions. Charts for calculating the rate of heat transfer by radiation between these gases and the surfaces of solids have been prepared by Prof H C Hottel. See *Trans Am Inst Chem Eng* vol 19 1927, and *Ind Eng Chem*, vol 19 p 888 1927.

[2] This definition is exact only if the surfaces are "gray" as explained in Sec 18.

the effective area of any surface is less than the actual area if the surface contains any concave portions. The *effective area* may be assumed equal to the area of the surface that would be obtained by filling up all the concave portions of the actual surface to make them flush with the adjacent plane or convex portions; and the emissivity of this imaginary surface may be
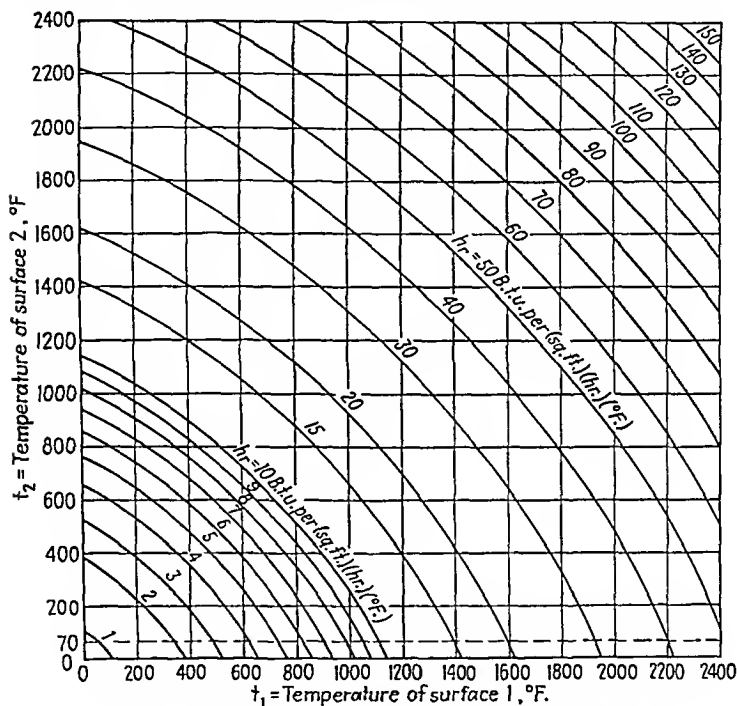

Fig. 10.—Radiation coefficient h_r, calculated by Eq. (21) and used in Eqs. (10) and (11).

taken as that of the actual surface increased toward unity in proportion to the depth and number of indentations.

c. Equations for Calculating the Net Rate of Heat Transfer by Radiation between Two Surfaces.—If two surfaces *1* and *2* are each uniform in temperature and are separated by a nonabsorbing medium (such as dry air), the net rate at which heat is transferred by radiation between the two surfaces can be calculated by the following equations:

Case 1 —If all the radiation from surface *1* is intercepted by surface *2* (*e g*, if surface *1* is completely inclosed by surface *2*, or if surface *1* is flat and surface *2* extends as far as the plane of surface *1*),

$$q = h_r A_1 (t_1 - t_2) F_e, \qquad (10)$$

where q = the net rate at which heat is transferred from surface *1* to surface *2* by radiation, B t u per hr

h_r = the radiation coefficient of heat transfer, determined from Fig 10 or calculated by Eq (21), B t u /(sq ft)(hr)(°F)

A_1 = the effective area of surface *1*, calculated as explained in the preceding paragraph, sq ft

t_1 and t_1 = the temperatures of surfaces *1* and *2*, °F

F_e = a factor to allow for the emissivities of the two surfaces, determined from Table 4

TABLE 4 —EMISSIVITY FACTOR F_e

e_1 and e_1 denote the emissivities of surfaces *1* and *2* and are evaluated from Table IV in the Appendix

No	Type of surface	F_e
1	Surface *1* the smaller of two concentric cylinders of radii r_1 and r_2 and of infinite length	$\dfrac{1}{\dfrac{1}{e_1} + \dfrac{r_1}{r_2}\left(\dfrac{1}{e_2} - 1\right)}$
2	Surface *1* the smaller of two concentric spheres of radii r_1 and r_2	$\dfrac{1}{\dfrac{1}{e_1} + \left(\dfrac{r_1}{r_2}\right)^2 \left(\dfrac{1}{e_2} - 1\right)}$
3	Surface *1* either of two infinite parallel planes	$\dfrac{1}{\dfrac{1}{e_1} + \dfrac{1}{e_2} - 1}$
4	Surface *1* of any shape but small compared to surface *2*	e_1
5	Surface *1* of any shape but almost as large as surface *2*	$\dfrac{1}{\dfrac{1}{e_1} + \dfrac{1}{e_2} - 1}$
6	Intermediate case between (4) and (5) above	$e_1 > F_e > \dfrac{1}{\dfrac{1}{e_1} + \dfrac{1}{e_2} - 1}$

Case 2 —If only part of the radiation from surface *1* is intercepted by surface *2* (i e, surface *2* does not completely inclose surface *1*),

$$q = h_r e_1 e_2 A_1 (t_1 - t_2) F_e, \qquad (11)$$

where q, h_r, A_1, t_1, and t_1 have the same meanings as in Eq (10), and

e_1 and e_2 = the emissivities of surfaces *1* and *2*, evaluated from Table IV in the Appendix.

F_A = the fraction of the total radiation from surface *1* which falls upon surface *2*, determined for several types of surfaces from Figs. 11 and 12.[1]
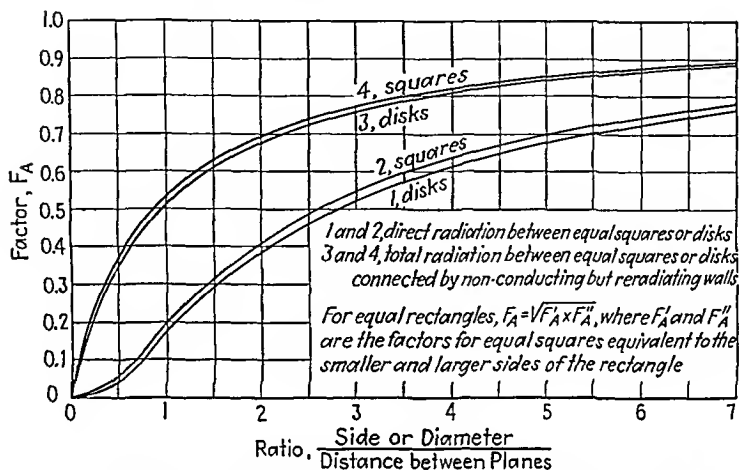

FIG. 11.—Radiation between equal squares or disks in parallel planes, directly opposed. (*H. C. Hottel.*)

This equation is not exact, but the approximation is satisfactory if e_1 and e_2 are 0.8 or higher.

It should be noted that the value of q calculated by Eq. (10) represents the net rate at which surface *1* gains or loses heat by radiation but that the value of q calculated by Eq. (11) represents only the net rate at which surface *1* gains or loses heat by radiation to surface *2* and does not include the heat gained or lost by radiation to other surfaces.

Illustrative Problem 1.—Calculate the rate at which heat is lost by radiation from a bare 3-in. nominal diameter wrought-iron pipe at 300°F. running through a room whose walls are at 70°F.

Solution.—Since the pipe is totally inclosed by the walls of the room, this problem is included in Case 1. From Fig. 10, h_r = 1.9 B.t.u./(sq. ft.)(hr.)(°F.); and from Table 4, type 4, $F_e = e_1$. From Table IV in the Appendix, the emissivity e_1 of dull, oxidized wrought iron is 0.94. Since

[1] Taken from Radiant Heat Transmission by H. C. Hottel, *Mech. Eng.*, vol. 52, p. 699, 1930.

the actual outside diameter of a 3-in pipe is 3 500 in , the heat loss due to radiation per foot of length is, by Eq (10),

$$q = 1\ 9 \times 3\ 14 \times \frac{3\ 500}{12} \times (300 - 70) \times 0\ 94$$

$$= 376 \text{ B t u per hr}$$

Illustrative Problem 2—Liquid oxygen is to be stored at atmosphere pressure in a metal Dewar flask consisting of a 12-in diameter inner sphere inclosed in a 13-in diameter outer sphere The spheres are made of brass, and the surfaces forming the vacuum space are highly polished   Calculate
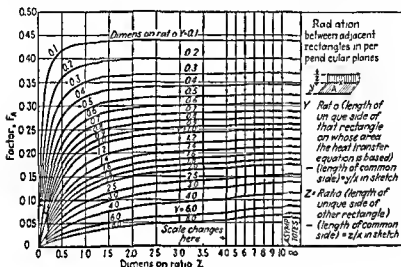
FIG 12—Radiation between adjacent rectangles in perpendicular planes (H C Hottel)

the rate at which the oxygen will be lost by evaporation if the temperature of the outer sphere will be 50°F (510°F abs), the temperature of the inner sphere will be −298°F (162°F abs), and the heat of vaporization of liquid oxygen is 91 8 B t u per lb

Solution—This problem is included in Case 1   By Eq (21),

$$h_r = \frac{0\ 173 \times 10^{-8} \times (162^4 - 510^4)}{(-298 - 50)}$$

$$= 0\ 332 \text{ B t u /(sq ft)(hr)(°F)}$$

The surface area of the inner sphere is 3 14 sq ft, and the factor F_e is obtained from Table 4, type 2   The emissivities of highly polished brass at the temperatures involved in the present problem are not given in Table IV in the Appendix   However, a conservative result should be obtained if a constant value of 0 028 is assumed   Hence, by Eq (10),

$$q = 0.332 \times 3.14 \times (-298 - 50) \left[\cfrac{1}{\dfrac{1}{0.028} + \left(\dfrac{12}{13}\right)^2 \left(\dfrac{1}{0.028} - 1\right)} \right]$$

$$= -5.56 \text{ B.t.u. per hr.},$$

the negative sign indicating that the direction of heat flow will be from the outer to the inner sphere. Assuming the heat transfer by convection to be negligible, the rate at which oxygen will be lost by evaporation is

$$\frac{5.56}{91.8} = 0.061 \text{ lb. per hr.}$$

Illustrative Problem 3.—Calculate the rate at which heat is lost by radiation through a 3-in. square peephole in a furnace wall 6 in. thick, the temperature within the furnace being 2200°F. and the surroundings being at 70°F.

Solution.—Since all the radiation entering the peephole from within the furnace must pass through an imaginary plane surface covering the inside end of the peephole, and since all the radiation lost through the peephole must pass through an imaginary plane surface covering the outside end, the problem is equivalent to one of calculating the rate of heat transfer by radiation between two parallel 3-in. squares 6 in. apart, one square being at 2200°F. and the other at 70°F. This problem is included in Case 2.

For $t_1 = 2200$°F. and $t_2 = 70$°F., from Fig. 10, $h_r = 40$ B.t.u./(sq. ft.)(hr.)(°F.). Since none of the radiation falling upon either of the squares is reflected, these imaginary surfaces are equivalent to black bodies and have emissivities equal to unity. Assuming that the walls of the peephole reradiate all the radiation falling upon them, from Fig. 11, curve 4, $F_A = 0.38$. Hence, the rate of heat loss due to radiation through the peephole is, by Eq. (11),

$$q = 40 \times 1 \times 1 \times \frac{3 \times 3}{144} \times (2200 - 70) \times 0.38$$
$$= 2020 \text{ B.t.u. per hr.}$$

Illustrative Problem 4.—A muffle-type furnace is 6 ft. wide, 12 ft. deep, and 6 ft. high. What fraction of the radiation emitted by the floor of the furnace falls (1) upon the side walls and (2) upon the top of the furnace? Provided they are well insulated, the side walls will reradiate all the heat that they receive after they have reached a state of equilibrium. (3) Under these conditions, what fraction of the heat emitted by the floor of the furnace will reach the top of the furnace?

Solution.—1. Referring to Fig. 12, for each of the 6- by 12-ft. side walls, $Y = \frac{6}{12}$ and $Z = \frac{6}{12}$, and therefore $F_A = 0.24$. For each of the 6- by 6-ft. side walls, $Y = \frac{12}{6}$ and $Z = \frac{6}{6}$, and therefore $F_A = 0.12$. Hence, the four side walls receive $2 \times (0.24 + 0.12) = 0.72$ of the radiation emitted by the floor.

2. Referring to Fig. 11, curve 2, F_A' for two 6-ft. squares 6 ft. apart is 0.20, and F_A'' for two 12-ft. squares 6 ft. apart is 0.41. Hence, the fraction F_A of the radiation emitted by the floor and received by the top of the

length at any given temperature to the maximum possible rate at which radiant energy of the same wave length can be emitted by any surface at the same temperature. This second definition follows from the first, since $I d\lambda$ is equal to the rate dq at which any surface emits radiant energy of wavelengths from λ to $\lambda + d\lambda$.

The *absorptivity* a of a surface for radiation of any wave length is the fraction of the incident radiation of the given wave length that the surface absorbs.

15. Kirchhoff's Law—Consider any two isolated, parallel plane surfaces that are both at the same temperature and that transmit none of the radiation incident upon them. Let q_λ denote *the maximum possible rate per unit area at which radiation of wave length λ can be emitted by any surface* at the given temperature and let e_1, e_2 and a_1, a_2 denote the emissivities and absorptivities of the two surfaces for radiation of this wave length. The rate per unit area at which each surface emits radiation of wave length λ is equal to $e q_\lambda$. Since the surfaces are parallel, all this radiation reaches the other surface, where the fraction a is absorbed[1] and the fraction $(1 - a)$ is reflected. The reflected radiation all returns to the first surface, and again the fraction a is absorbed, and the fraction $(1 - a)$ is reflected. Thus, part of the radiation emitted by each surface undergoes repeated reflections, the fraction a being absorbed and the fraction $(1 - a)$ being reflected each time that the radiation falls upon either of the surfaces.

Since the two surfaces are isolated and are at the same temperature, the *net* rate at which each loses heat by radiation is zero. Hence, the rate $e q_\lambda$ at which each surface emits radiation of wave length λ is equal to the sum of (1) the rate at which it absorbs the radiation of wave length λ emitted by the other surface plus (2) the rate at which it absorbs the radiation of wave length λ returned to it by reflection. These two equations are readily reduced to the single equation

$$\frac{e_1}{a_1} = \frac{e_2}{a_2} \tag{12}$$

---

[1] The emissivities and absorptivities of the two surfaces are assumed independent of the angles of emission and incidence. The assumption that the emissivity is independent of the angle of emission is known as *Lambert's cosine principle*.

This result is known as *Kirchhoff's law* and may be stated, "For any given wave length and temperature, the ratio of the emissivity to the absorptivity is a constant for all surfaces."

16. Black Body.—A *black body* is defined as one that absorbs all the radiation of all wave lengths falling upon it. From this definition it follows that the absorptivity a of a black body is unity at all wave lengths and temperatures. No actual surfaces are black in this sense, although the imaginary surface covering a small opening in an otherwise completely enclosed box is approximately so, since all the radiation falling upon it enters the box and undergoes successive reflections until completely absorbed.

17. Equivalence of Emissivity and Absorptivity.—Kirchhoff's law states in effect that any surface emits radiation in proportion to its ability to absorb radiation. It follows that a black body, which absorbs a maximum fraction of the radiation of all wave lengths incident upon it (*i.e.*, all of it), must also emit radiation of all wave lengths at the maximum possible rate. Hence, from the second definition of emissivity given in Sec. 14, it follows that the emissivity e of a black body is unity at all wave lengths. Since the emissivity and the absorptivity of a black body are equal at all wave lengths and temperatures, it follows from Kirchhoff's law that *the emissivity and the absorptivity of any surface are equal at the same wave length and temperature.* Thus, at the same wave length and temperature, for any surface,

$$e = a. \tag{13}$$

18. Gray Surfaces.—A *gray surface* is defined as one whose emissivity e is the same at all wave lengths and temperatures. Hence, the ratio of the rate at which a gray surface emits radiation of any wave length to the rate at which a black body at the same temperature would emit radiation of the given wave length is constant for all wave lengths. Therefore, *for a gray surface,*

$e =$ (the *total* rate at which a gray surface emits radiation of all wave lengths) $\div$ (the *total* rate at which a black body at the same temperature would emit radiation of all wave lengths).[1]   (14)

[1] Applied to any actual surface, the quantity defined in this way is called the *total*, or *hemispherical*, *emissivity*.

By Eq (13) the absorptivity a of a gray surface is also the same at all wave lengths and temperatures Therefore, *for a gray surface*,

a = (the *total* rate at which a gray surface absorbs the radiation of all wave lengths incident upon it) − (the *total* rate at which a black body would absorb the same radiation) (15)

The definition of emissivity given in Sec 12a is based on Eqs (13), (14) and (15) and is therefore strictly valid only for gray surfaces None of the surfaces encountered in engineering work is actually "gray," but the error introduced by assuming them to be such is usually small

19 Stefan-Boltzmann Equation —The *total* rate at which a black body at any temperature emits radiation of all wave lengths is represented by the area under the intensity wave-length curve for a black body (Fig 13) and can be calculated by the Stefan Boltzmann equation

$$q = 0.173 \times 10^{-8} A T^4, \qquad (16)$$

where q = the total rate at which a black body emits radiation of all wave lengths B t u per hr

A = the area of the black body sq ft

T = the absolute temperature of the black body, °F abs

This equation was first established by Stefan on the basis of the experimental work of Dulong and Petit and was subsequently deduced by Boltzmann from purely theoretical considerations

It follows from Eqs (14) and (16) that the total rate at which a gray surface at any temperature emits radiation of all wave lengths is given by the equation

$$q = 0.173 \times 10^{-8} e A T^4 \qquad (17)$$

where e represents the emissivity of the gray surface

20 Net Rate of Heat Transfer by Radiation between Two Surfaces —If the surfaces are separated by some nonabsorbing medium the net rate at which any surface 1 loses heat by radiation to some other surface 2 is equal to (1) the rate at which surface 1 emits radiation (2) minus the rate at which it reabsorbs the fraction of this radiation that is reflected back to it by surface 2, and (3) minus the rate at which it absorbs the radiation emitted by surface 2

Case 1.—Consider first the case where all the radiation emitted by surface *1* falls upon surface *2*. The simplest problem of this type is that of a gray surface at the uniform absolute temperature T_1 completely inclosed by a black body at the uniform absolute temperature T_2. Letting A_1 denote the area and e_1 the emissivity of the gray surface, by Eq. (17) the rate at which the gray surface *emits* radiation is $0.173 \times 10^{-8} e_1 A_1 T_1^4$. Since all this radiation is absorbed by the black body, item (2) in the preceding paragraph is zero. The rate at which the gray surface absorbs the radiation emitted by the black body can be determined as follows: If the gray surface were replaced by a black body of the same size as the gray surface but at the temperature T_2, such a body would be in equilibrium with its surroundings and would therefore emit and absorb radiation at the same rate. This rate, by Eq. (16), is equal to $0.173 \times 10^{-8} A_1 T_2^4$ and is the rate at which radiation falls upon the gray surface. Since absorptivity and emissivity are numerically equal, by Eq. (15) the rate at which the gray surface *absorbs* radiation is

$$0.173 \times 10^{-8} e_1 A_1 T_2^4.$$

Hence, the net rate q at which the gray surface loses heat by radiation is

$$q = 0.173 \times 10^{-8} e_1 A_1 (T_1^4 - T_2^4). \tag{18}$$

If surface *2* is gray instead of black, part of the radiation that reaches it from surface *1* is reflected and returned to surface *1*. Hence, the net rate at which surface *1* loses heat by radiation depends not only upon the emissivities of both surfaces but also upon the shapes and relative positions of the two surfaces. Assuming that the reflection is diffuse rather than specular, it can be shown[1] that the net rate at which the smaller of two concentric cylinders of infinite length loses heat by radiation is

$$q = 0.173 \times 10^{-8} A_1 \left[\cfrac{1}{\cfrac{1}{e_1} + \cfrac{r_1}{r_2}\left(\cfrac{1}{e_2} - 1\right)} \right] (T_1^4 - T_2^4), \tag{19}$$

and the net rate at which the smaller of two concentric spheres loses heat by radiation is

$$q = 0.173 \times 10^{-8} A_1 \left[\cfrac{1}{\cfrac{1}{e_1} + \left(\cfrac{r_1}{r_2}\right)^2\left(\cfrac{1}{e_2} - 1\right)} \right] (T_1^4 - T_2^4), \tag{20}$$

[1] *Proc. Phys. Soc.*, vol. 41, p. 569, 1929.

where the subscript 1 indicates the smaller of the two cylinders or spheres r denotes radius and the other symbols have the same meanings as in Eq (18) The term in brackets in these two equations is the factor F_a listed in Table 4 the last four cases in this table being special cases of the first two

The radiation coefficient of heat transfer h given in Fig 10 is defined by the equation

$$h_r = \frac{0\ 173 \times 10^{-8}(T_1^4 - T_2^4)}{(t_1 - t_2)} \tag{21}$$

Equation (10) is obtained by combining Eq (21) with either Eq (19) or Eq (20) and replacing the term in brackets with F_a

Case 2—Consider next the case where only part of the radiation emitted by surface *1* falls upon surface *2* Letting f denote the fraction of the radiation that is emitted from an element dA of surface *1* and that falls upon surface *2* and remembering that surface *2* absorbs the fraction e_2 of the radiation falling on it the rate at which surface *2* absorbs radiation from dA is $0\ 173 \times 10^{-8} e_1 e_2\, dA\, T_1^4 f$ The rate at which radiation from surface *2* falls upon dA is $0\ 173 \times 10^{-8} e_1\, dA\, T_2^4 f$ since this is the rate at which an element of surface *2* placed in the position of dA would emit radiation to or absorb radiation from surface *2* Hence the rate at which dA absorbs radiation from surface *2* is $0\ 173 \times 10^{-8} e_1 e_2\, d\, t\, T_2^4 f$ Neglecting the rates at which the radiation reflected back to each surface is reabsorbed (permissible if e_1 and e_2 are both large) the net rate at which heat is transferred from $d\, 1$ to surface *2* by radiation is

$$0\ 173 \times 10^{-8} e_1 e_2\, d\, 1(T_1^4 - T_2^4)f$$

and the net rate q at which heat is transferred from the entire surface *1* to surface *2* is

$$q = 0\ 173 \times 10^{-8} e_1 e_2\, 1_1(T_1^4 - T_2^4)F_A \tag{22}$$

where F_A is the average of the values of f for each element of surface *1* Equation (11) is obtained by combining this equation with Eq (21)

CHAPTER III

CONVECTION

21. Introduction.—Convection is the transfer of heat from one part of a fluid to another by the mixing of the warmer particles of the fluid with the cooler. The motion of the fluid may be caused either by differences in density within the fluid or by some mechanical device, such as a stirrer, pump, or fan. The first process is called *natural convection;* and the second, *forced convection.*
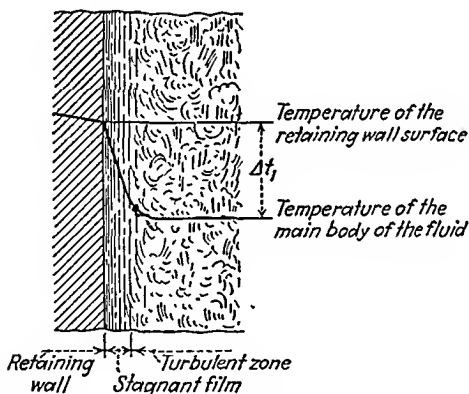

Fig. 14.—Temperature gradient through a fluid in contact with a warmer retaining wall.

Consider the case of a fluid in contact with a retaining wall whose temperature is higher than that of the fluid, as shown in Fig. 14. Although the main body of the fluid is likely to be in turbulent motion, the thin film of fluid in contact with the retaining wall is relatively stagnant as the result of friction with the wall. The thickness of this stagnant film is not clearly defined and varies as the degree of turbulence of the adjacent fluid varies. Since heat is transferred through this film largely by conduction rather than by convection, the entire process by which heat is transferred from the retaining wall to the main

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body of the fluid is rather complicated Calculations are simplified, however, by the use of so-called *convection coefficients* which include the combined effect of both conduction and convection

In this and the following chapter, it is assumed that a state of equilibrium has been reached, *i e*, the temperature gradient is assumed to remain constant and not change with time, and the rate of heat transfer through the retaining wall is assumed equal to the rate at which heat is absorbed or given up by the fluid in contact with the wall In addition, it is assumed that radiation is either negligible or else is taken into account by the method given in Chap II In this connection, the rate of heat transfer by radiation is usually small compared to the rate of heat transfer by convection An important exception is that of heat transfer by natural convection from any surface to a gas when the surface is exposed to other surfaces at higher or lower temperatures For example, a bare steam pipe may lose more heat by radiation to the walls of a room than it loses by natural convection to the surrounding air

22 Definition of the Film Coefficient h —The *film coefficient* of convection h is defined as the rate of heat transfer between the retaining wall and the fluid per unit area of the retaining wall and per degree temperature difference between the surface of the wall and the main body of the fluid From this definition it follows that

$$q = hA(\Delta t_i), \qquad (23)$$

where q = the rate of heat transfer by convection, B t u per hr

h = the film coefficient, B t u /(sq ft)(hr)(°F)

A = the area of the retaining wall, sq ft

Δt_i = the temperature difference between the surface of the wall and the main body of the fluid °F

The quantity h is also called the *individual coefficient* or the *surface coefficient*

23 Factors Affecting the Film Coefficient h —There are three types of heat transfer by convection (1) heat transfer in which the fluid does not change phase (*i e*, the fluid remains either a liquid or a gas throughout the process), (2) heat transfer in which a vapor is condensed and (3) heat transfer in which a liquid is evaporated The film coefficients for all three types depend not only upon the kind of fluid involved and upon the pressure

and temperature of the fluid but also upon the type of retaining wall surface (*e.g.*, flat surfaces, the inside surface of tubes, or the outside surface of tubes) and upon the position of the retaining wall surface (*e.g.*, horizontal or vertical). In addition, the film coefficients for each individual type of convection depend upon the following factors:

1. For the first type in which the fluid undergoes no change of phase, the film coefficient depends upon the turbulence of the fluid and increases as the turbulence increases. In the case of natural convection, the turbulence, and consequently the film coefficient, increases as the temperature difference between the main body of the fluid and the surface of the retaining wall increases. In the case of flow through tubes, regardless of whether the motion is the result of natural or forced convection, the film coefficient increases as the velocity of the fluid increases. Although the type of flow depends also upon the diameter of the tube and upon the density and viscosity of the fluid,[1] at low velocities the flow is likely to be *streamline* (also called *viscous* or *laminar* flow), and at higher velocities it is usually *turbulent*. The transition from streamline to turbulent flow is accompanied by a considerable increase in the film coefficient. In the case of flow outside tubes, the film coefficient depends also upon the direction of flow. For any given velocity, the film coefficient is usually higher for flow normal to the tubes than for flow parallel to the tubes.

The film coefficient depends also upon whether the fluid is being heated or cooled, because the temperature of the film of fluid in contact with the surface of the retaining wall is at a higher temperature than the main body of the fluid when the fluid is being heated and lower when cooled. The difference in the film coefficients for heating and for cooling decreases as the turbulence of the fluid increases and in the case of gases is usually negligible.

2. For the second type of convection in which a vapor is condensed, the film coefficient depends upon whether the condensation is *dropwise* or *filmwise*. As the names imply, with dropwise

[1] For isothermal flow inside straight tubes, for example, the flow is usually streamline if the product $DV\rho/\mu$ [D = inside tube diameter, ft.; V = velocity, ft. per sec.; ρ = density, lb. per cu. ft.; and μ = viscosity, lb./(ft.)(sec.)] is less than 2,100 and is turbulent, if greater.

condensation the condensate collects on the surface of the retaining wall in drops, with filmwise condensation, the condensate forms a continuous film over the entire surface   Although film coefficients for dropwise condensation are four to eight times as high as for filmwise, the latter type is the one more likely to occur in practice, since dropwise condensation usually takes place only on polished surfaces

In the case of filmwise condensation, the film of condensate adhering to the surface of the retaining wall acts somewhat as insulation, and consequently the film coefficient depends upon both the thickness and the turbulence of this film   For example, if condensate is allowed to drip on to a horizontal tube from other tubes above it, the film of condensate will be thicker, and therefore the film coefficient will be smaller   Similarly, since all the condensate formed on a vertical surface must flow down the surface, the average thickness of the film of condensate tends to increase as the height of the surface is increased   However, up to a certain rate of condensation the flow of condensate down the surface is streamline, and for greater rates it is turbulent   As long as the flow is streamline, the average film coefficient decreases as the height of the surface is increased, but if the flow is turbulent, both the turbulence and the film coefficient increase as the height is increased

The film coefficient is also affected by the velocity of the vapor If the vapor strikes the surface of the retaining wall with a velocity sufficient to sweep away part of the film of condensate, the thickness of the film is reduced and the film coefficient is increased

3 The various factors affecting the film coefficient for the third type of convection in which a liquid is evaporated are not yet so well established as are the factors affecting the film coefficients for the other two types   It is known, however, that the film coefficient depends largely upon how great a fraction of the heating surface is in contact with the liquid rather than with the vapor bubbles   Since the rate of heat transfer from the heating surface to the liquid is much higher than the rate of heat transfer to the vapor the film coefficient is increased by anything that either reduces the size of the vapor bubbles formed on the heating surface or reduces the time required for the bubbles to break away   The size of the bubbles is partially

determined by the tendency of the liquid to wet the heating surface. Thus, the film coefficient is higher for a liquid having a high tendency to wet the surface than for one having a low tendency. The time required for the bubbles to break away from the heating surface is greatly affected by the turbulence of the liquid. If the motion of the liquid is entirely the result of natural convection, both the turbulence and the film coefficient increase as the temperature difference between the heating surface and the main body of the fluid increases up to a certain critical value. As the temperature difference increases beyond this critical value, however, the film coefficient decreases very rapidly, because the heating surface becomes completely separated from the liquid by a film of vapor. If the liquid is violently agitated by some form of mechanical stirrer, the temperature difference has very little effect on the film coefficient. Finally, the film coefficient is affected by the condition of the heating surface and is higher if the surface is rough instead of smooth.

In addition to the various factors already mentioned, the film coefficients for all three types of heat transfer by convection are affected by the presence of scale or other deposits on the surface of the retaining wall. Such deposits act as insulation and may greatly reduce the rate of heat transfer. Allowance for them may be made by the method explained in Sec. 26.

Values of the film coefficient must be determined experimentally for most cases, because the motion of the fluid and the temperature distribution are too complicated to permit these coefficients to be predicted from purely theoretical considerations. Fortunately, the experimental results obtained with a few fluids can be used to predict film coefficients for other fluids by means of dimensional analysis (see Secs. 29 and 30). Dimensionless equations, suggested by dimensional analysis and representing with reasonable accuracy the results of numerous investigators, are available in the technical literature. Based on certain of these equations, charts and tables have been prepared which can be used to determine film coefficients for various fluids under various conditions. These charts and tables, together with the dimensionless equations upon which they are based, are given in Chap. IV.

24. Definition of the Over-all Coefficient U.—In most industrial heat-transfer equipment, one fluid receives heat from

another fluid, the two fluids being separated by a solid retaining wall In designing such equipment, the temperatures at which the two fluids are to enter and leave the equipment are usually known, but the temperature of the retaining wall is not known Consequently, the expected rate of heat transfer cannot be calculated directly from the film coefficients, since these are based on the temperature difference between the retaining wall and the fluids However, the expected rate of heat transfer can be readily calculated from the over-all coefficient U, because this coefficient is based on the temperature difference between the two fluids

The *over-all coefficient* U is defined as the rate at which heat is transferred from one fluid to another per unit area of the retaining wall and per degree temperature difference between the two fluids From this definition, it follows that

$$q = UA(\Delta t), \qquad (24)$$

where q = the rate of heat transfer by convection, B t u per hr
  U = the over-all coefficient, B t u /(sq ft)(hr)(°F)
  A = the area of the retaining wall, sq ft
  Δt = the temperature difference between the main body of one fluid and the main body of the other, °F

If the temperature difference between the two fluids is not constant throughout the equipment, an average value must be used for Δt The correct method of calculating this average value is discussed in Sec 27   If the heat transfer surface consists of tubes, the value of the area A to be used depends upon whether the coefficient U is based on the inside or outside area of the tubes, as explained in the next section

25 Relation between the Over-all Coefficient and the Film Coefficients —The over-all coefficient U can be calculated from the film coefficients when these are known for both sides of the retaining wall

For *flat* or slightly curved walls free from scale,

$$\frac{1}{U} = \frac{1}{h_1} + \frac{L}{k} + \frac{1}{h_3}, \qquad (25)$$

where U = the over all coefficient, B t u /(sq ft)(hr)(°F)
  h_1 = the film coefficient for one side of the wall, B t u /(sq ft)(hr)(°F)

h_2 = the film coefficient for the other side of the wall, B.t.u./(sq. ft.)(hr.)(°F.).

L = the thickness of the wall, ft.

k = the thermal conductivity of the wall, B.t.u./(ft.)(hr.)(°F.).

For *tubes*, the coefficient U may be based on either the inside or the outside area of the tubes. For ordinary thin-walled tubes free from scale, the over-all coefficient U may be calculated by the following approximate equations: Based on the inside area of the tube,

$$\frac{1}{U_1} = \frac{1}{h_1} + \frac{L}{k} + \frac{D_1}{h_2 D_2}, \tag{26}$$

and based on the outside area of the tube,

$$\frac{1}{U_2} = \frac{D_2}{h_1 D_1} + \frac{L}{k} + \frac{1}{h_2}, \tag{27}$$

where U_1 and U_2 = the over-all coefficients based on the inside and the outside tube areas, respectively, B.t.u./(sq. ft.)(hr.)(°F.).

h_1 and h_2 = the film coefficients for the inside and outside tube surfaces, respectively, B.t.u./(sq. ft.)(hr.)(°F.).

D_1 and D_2 = the inside and outside diameters of the tube, respectively, ft. or in.

L = the thickness of the tube wall, ft.

k = the thermal conductivity of the tube wall, B.t.u./(ft.)(hr.)(°F.).

For tubes having unusually heavy walls, the L/k terms in these two equations should be replaced by $(D_1/2k) \log_e (D_2/D_1)$ and $(D_2/2k) \log_e (D_2/D_1)$, respectively.

For practical purposes, sufficiently accurate values of the over-all coefficient U for both flat walls or thin-walled tubes free from scale can frequently be calculated by the following approximate equation:

$$\frac{1}{U} = \frac{1}{h_1} + \frac{1}{h_2}, \tag{28}$$

where h_1 and h_2 are the film coefficients for the two sides of the wall. This equation can be quickly solved by means of Fig. 15.

Equation (25) can be derived as follows The rates of heat transfer between the two fluids and the surfaces of the retaining
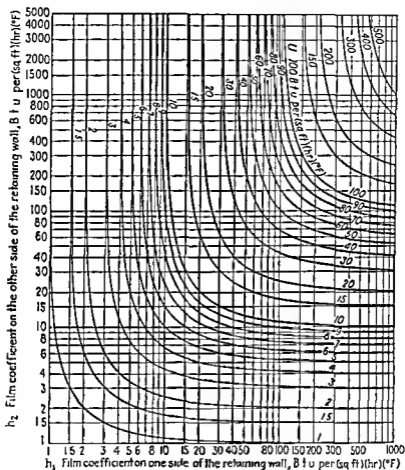
Fig 15—Chart for determining approximate values of the over all coefficient U from the film coefficients h_1 and h_2 [based on Eq (28)]

wall are, by Eq (23),

$$q = h_1 A (\Delta t_1)$$

and

$$q = h_2 A (\Delta t_2)$$

where Δt_1 and Δt_2 are the temperature differences between the main body of each fluid and the surface of the wall By Eq (3)

in Chap. I the rate of heat transfer by conduction through the wall itself is

$$q = \frac{kA(\Delta t_w)}{L},$$

where Δt_w is the temperature difference between the two surfaces of the wall. By Eq. (24) the rate of heat transfer from one fluid to the other is

$$q = UA(\Delta t),$$

where Δt is the temperature difference between the main body of one fluid and the main body of the other. Since it is assumed that a state of equilibrium has been reached, the rate of heat transfer q is the same in all of these equations. Equation (25) is obtained by solving each of the equations for the temperature difference term and substituting these expressions into the equation

$$\Delta t = \Delta t_1 + \Delta t_w + \Delta t_2.$$

Equations (26) and (27) are obtained in a similar manner. These equations are based on the simplifying assumption that the rate of heat transfer through the tube wall itself can be calculated with sufficient accuracy by Eq. (3) in Chap. I. This assumption is permissible for ordinary thin-walled tubes, but for tubes having heavy walls the rate of heat transfer through the tube wall must be calculated by Eq. (4) in Chap. I.

Equation (28) is approximately equivalent to Eqs. (25), (26), and (27) if the retaining wall is thin and is made of some material having a high thermal conductivity, because for these conditions the L/k terms in these equations become negligible and the D_1/D_2 ratio is approximately unity.

26. Effect of Scale on the Over-all Coefficient U.—The equations for calculating the over-all coefficient U given in the preceding section involve the assumption that the surfaces of the retaining wall are free from scale. If scale is present on either surface, the over-all coefficient U can be calculated by the equation

$$\frac{1}{U_s} = \frac{1}{U_c} + \frac{1}{h_s}, \tag{29}$$

where U_s = the over-all coefficient when one surface of the retaining wall is covered with scale, B.t.u./(sq. ft.)(hr.)(°F.).

U_c = the over all coefficient when both surfaces of the retaining wall are clean, B t u /(sq ft)(hr)(°F)

h_s = the *scale coefficient* B t u /(sq ft)(hr)(°F) which is defined as the rate of heat transfer through the scale per unit area of scaly surface and per degree temperature drop through the scale

If scale is present on both surfaces, a second $1/h_s$ term must be added to the equation

Values of the scale coefficient h_s can be calculated if the thermal conductivity and the thickness of the scale are known   In
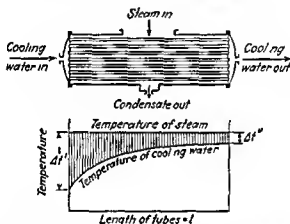
Fig 16 —Variation of the temperature difference in a condenser

practice however it is more satisfactory to determine h_s from values of the over all coefficient U measured before and after the surfaces of the retaining wall have become covered with scale   A table of values of h_s is given in Chap IV

If a $1/h_s$ term is added to the right-hand members of Eqs (25) (26) (27) and (28) these equations become equivalent to Eq (29)   Equation (29) can be derived in a manner analogous to that used in deriving these equations

27 Mean Temperature Difference

—In general the temperature difference between two fluids flowing through any kind of heat-transfer equipment does not remain constant throughout the equipment   For example consider a steam condenser in which the cooling water makes a single pass through the tubes as shown in Fig 16   The temperature difference between the steam and the cooling water is greatest at the section where the water enters

exchangers in which the **two fluids flow** at right angles to each other, the true mean temperature difference can be determined by
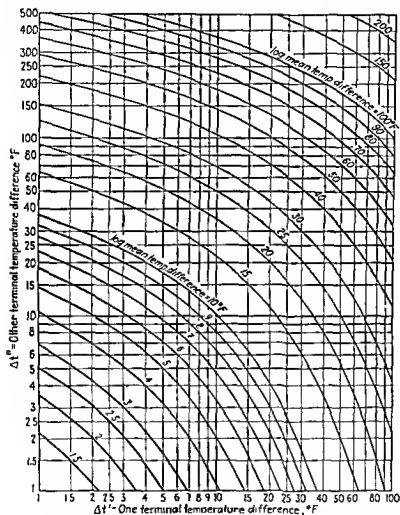
Fig 17—Chart for determining the logarithmic mean temperature difference from the terminal temperature differences

multiplying the logarithmic-mean temperature difference *calculated for counterflow* by a correction factor F obtained from Fig 19, also prepared by **Bowman**, Mueller, and Nagle. The nomenclature used in these figures is defined in Table 5

If the temperature of one of the fluids remains constant, as shown in Fig. 16, Eq. (30) can be derived as follows: Let t, c_p, and w denote the temperature, specific heat, and weight rate
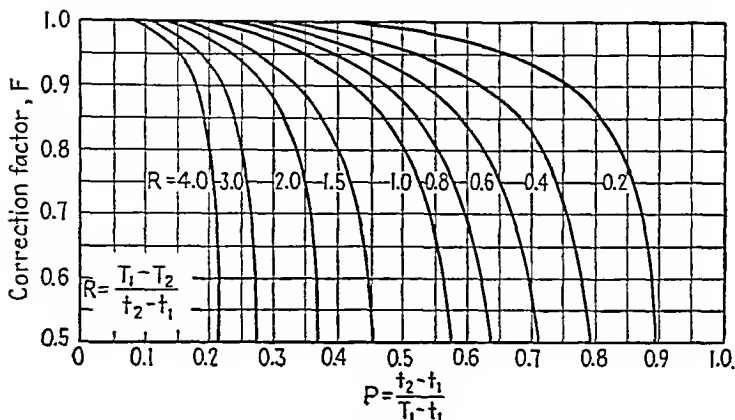
FIG. 18a.—Correction factor for multipass heat exchangers with *one* shell pass and two, four, or any multiple of tube passes. (*Bowman, Mueller, and Nagle.*)
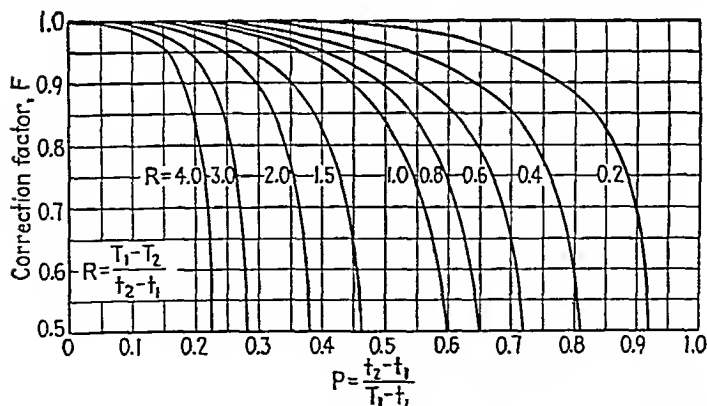
FIG. 18b.—Correction factor for multipass heat exchangers with *one* shell pass and three, six, or any multiple of tube passes, majority of tube passes in counterflow. (*Bowman, Mueller, and Nagle.*)

of flow, respectively, of the fluid whose temperature changes; let A' denote the total heat-transfer surface per foot of length of the tube bundle; and let l denote the length of the tube bundle. If the equipment is perfectly insulated, the rate of heat transfer

through a section of the tube bundle of length dl is equal to the heat absorbed per hour by the fluid flowing through this section
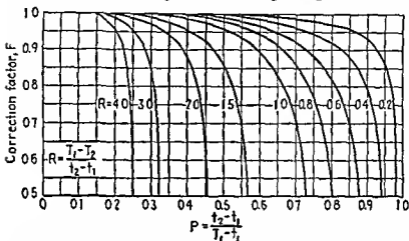
Fig. 18c —Correction factor for multipass heat exchangers with *two shell passes and four, eight or any multiple of tube passes* (*Bowman Mueller and Nagle*)
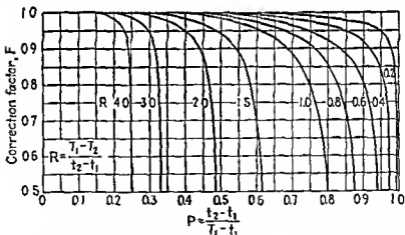
Fig. 18d —Correction factor for *multipass* heat exchangers with *three shell passes and six, twelve or any multiple of tube passes* (*Bowman Mueller and Nagle*)

Hence,

$$U(A'dl)(\Delta t) = wc_p\, dt = -wc_p d(\Delta t),$$

or

$$dl = -\frac{wc_p}{UA'}\frac{d(\Delta t)}{(\Delta t)}.$$

If the wc_p/UA' term remains constant, the following equation is obtained on integrating between the limits $l = 0$, $\Delta t = \Delta t'$ and $l = l$, $\Delta t = \Delta t''$:

$$l = \frac{wc_p}{UA'} \cdot \ln\left(\frac{\Delta t'}{\Delta t''}\right).$$

The mean temperature difference for the entire tube bundle is
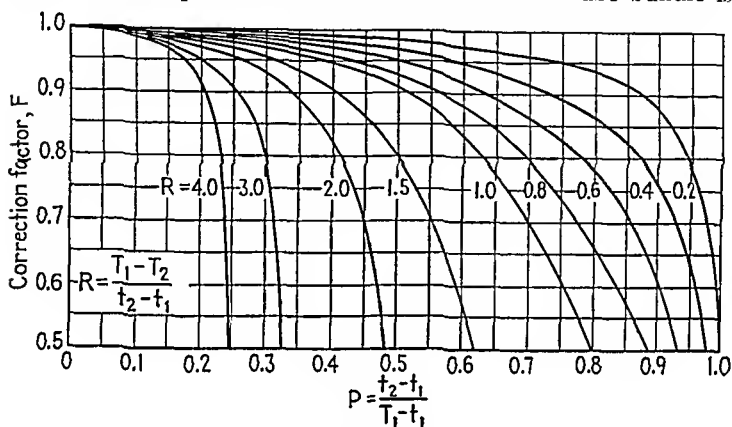
FIG. 19.—Correction factor for single-pass crossflow heat exchangers. (*Bowman, Mueller, and Nagle.*)

equal to the mean height of the shaded area in Fig. 16, or

$$\Delta t_{\text{mean}} = \frac{\text{shaded area}}{\text{length of shaded area}} = \frac{\int(\Delta t)dl}{l}.$$

Equation (30) is obtained by inserting the preceding expressions for l and dl into this equation and integrating between the limits $\Delta t = \Delta t'$ and $\Delta t = \Delta t''$.

TABLE 5.—NOMENCLATURE FOR FIGS. 18 AND 19

T_1	Inlet temperature of shell-side fluid, °F.
T_2	Outlet temperature of shell-side fluid, °F.
t_1	Inlet temperature of tube-side fluid, °F.
t_2	Outlet temperature of tube-side fluid, °F.
P	Dimensionless ratio = $(t_2 - t_1)/(T_1 - t_1)$.
R	Dimensionless ratio = $(T_1 - T_2)/(t_2 - t_1)$.
F	Correction factor by which the logarithmic-mean temperature difference *for counterflow* is multiplied to obtain the true mean temperature difference.
$(T_1 - t_2)$	Terminal temperature difference for counterflow, °F.
$(T_2 - t_1)$	Terminal temperature difference for counterflow, °F.

It can be shown in a somewhat similar manner that Eq (30) also applies when the temperatures of both fluids vary, provided that the equipment is insulated and the quantity wc_p/UA' is constant for each fluid

28 Design of Equipment Involving Heat Transfer by Convection —The amount of heat transfer surface required in any equipment in which heat is to be transferred by convection from one fluid to another can be calculated by Eq (24)

$$q = UA(\Delta t) \qquad (24)$$

It is first necessary to know the rate q at which heat is to be transferred from one fluid to the other, (2) the over-all coefficient of convection U, and (3) the *mean* temperature difference Δt between the two fluids These three quantities can be calculated as follows

1 If the equipment is to be insulated, the rate of heat transfer q can be calculated from the properties of either fluid, since the rate at which the warmer fluid will give up heat must equal the rate at which the cooler fluid will absorb heat If the fluid used in the calculation does not change phase, the rate of heat transfer q can be calculated from the rate of flow, the specific heat, and the entering and leaving temperatures of the fluid If the fluid evaporates or condenses the rate of heat transfer q can be calculated from the rate of flow and the heat of vaporization of the fluid However, any heat given up or absorbed by the fluid as the result of its being superheated or subcooled must also be included

It may be noted that the temperatures and the rates of flow of the two fluids may not all be chosen arbitrarily, since (*a*) the values must be such that the heat balance is satisfied, and (*b*) at every section of the equipment the temperature of the warmer fluid must be higher than the temperature of the cooler fluid For example, if neither fluid changes phase and if the entering and leaving temperatures of the two fluids are specified the rate of flow of either of the fluids may be assigned any value desired, *but the rate of flow of the other fluid must then be determined from the heat balance*

2 The over-all coefficient U can be calculated from the film coefficients for the two fluids and these can be determined from the charts and tables given in Chap IV The method of using these charts and tables is explained in Sec 32

The film coefficients for all types of convection depend upon the temperature of the fluid. If the fluid does not change phase, the temperature of the fluid changes as the fluid flows through the equipment, and consequently the film coefficient also changes. A sufficiently accurate average value can usually be obtained, however, if the film coefficient is evaluated at the *average temperature* of the fluid, the average temperature being taken as the arithmetic average of the temperatures at which the fluid enters and leaves the equipment.

If the heat-transfer surface is to consist of tubes, it may be necessary to select tentative values for the size and the number of the tubes before the film coefficients can be determined. For example, the film coefficient for the fluid flowing through the tubes usually depends upon the velocity of the fluid, which cannot be calculated until the size and the number of the tubes have been selected. If these values are selected in advance, only the length of the tubes remains to be calculated after the amount of heat-transfer surface required has been determined by Eq. (24). If the tube length turns out to be either too long or too short for practical purposes, a new value for either the size or the number of the tubes must be selected, and the calculations repeated.

Having determined the film coefficients for the two fluids, the over-all coefficient U can be calculated by the equations given in Sec. 25, or it can be evaluated approximately by Fig. 15. Allowance for scale can be made by Eq. (29).

3. The *mean* temperature difference between the two fluids must be used for Δt, and this can be calculated as explained in Sec. 27.

Problems illustrating these calculations are given in the next chapter.

CHAPTER IV

CONVECTION COEFFICIENTS

29 Introduction—Charts and tables for determining film coefficients for various kinds of convection are given in this chapter Charts and tables for determining over-all coefficients are not given, since these coefficients can be readily calculated from the film coefficients by means of the equations given in Sec 25 of Chap III

Experimentally, film coefficients are more difficult to measure than over-all coefficients, because the temperature of the retaining wall must be measured in addition to the temperatures of the fluids However, this increased experimental difficulty is compensated for by the fact that since film coefficients depend upon fewer variables, the results are more easily correlated Regarding the correlation of experimental data, the problem of correlation can frequently be simplified by means of dimensional analysis Dimensional analysis may not only indicate how the number of variables can be reduced but may also permit the results obtained with one or two fluids to be used to predict values for other fluids The simplification sometimes made possible is illustrated by the example given in the next section

30 An Illustration of the Use of Dimensional Analysis—Consider a sphere submerged in a liquid and rolling down an incline as shown in Fig 20 The sphere accelerates until the resistance offered by the liquid exactly balances the force F acting on it in the direction of its motion Let V denote the maximum velocity attained by the sphere

For any given diameter sphere kind of liquid and liquid temperature the maximum velocity V corresponding to different values of the force F could be determined experimentally by changing the angle of inclination α If the resulting values of V were plotted against F a separate curve would be obtained for each combination of diameter kind of liquid and liquid temperature used On the other hand if the results were plotted on the coordinates suggested by dimensional analysis all the data would fall upon a *single* curve

The maximum velocity V evidently depends upon the force F the diameter of the sphere D and the density ρ and viscosity μ of the liquid, i e

$$V = f(F, D, \rho, \mu)$$

This unknown function can be expressed as an exponential series,

$$V = a_1 F^{w_1} D^{x_1} \rho^{y_1} \mu^{z_1} + a_2 F^{w_2} D^{x_2} \rho^{y_2} \mu^{z_2} + \cdots \tag{31}$$

in which each term is of the form $aF^w D^x \rho^y \mu^z$, and a, w, x, y, and z are dimensionless constants. *Each of the terms in the right-hand member of this equation must have the same net dimensions as those of the left-hand member.*

In terms of the three fundamental dimensions mass m, length l, and time θ, the dimensions of the velocity V are l/θ, of the force F are ml/θ^2 (mass $\times$ acceleration), of the diameter D are l, of the density ρ are m/l^3, and
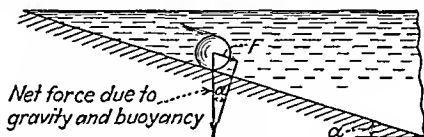
FIG. 20.—Sphere submerged in a liquid and rolling down an incline.

of the viscosity μ are $m/l\theta$ (force $\times$ distance/area $\times$ velocity). Hence, equating the dimensions of the general term $aF^w D^x \rho^y \mu^z$ to the dimensions of V,

$$\left(\frac{ml}{\theta^2}\right)^w (l)^x \left(\frac{m}{l^3}\right)^y \left(\frac{m}{l\theta}\right)^z = \frac{l}{\theta},$$

or

$$m^{w+y+z} l^{w+x-3y-z-1} \theta^{-2w-z+1} = 1.$$

This equation will be satisfied if each of the exponents is equal to zero:

$$\begin{aligned} w + y + z &= 0, \\ w + x - 3y - z - 1 &= 0, \\ -2w - z + 1 &= 0, \end{aligned}$$

Solving these three simultaneous equations for x, y, and z in terms of w,

$$\begin{aligned} x &= -1, \\ y &= w - 1, \\ z &= -2w + 1. \end{aligned}$$

Hence, each term of the series must be of the form

$$aF^w D^x \rho^y \mu^z = aF^w D^{-1} \rho^{w-1} \mu^{-2w+1} = a\left(\frac{\mu}{D\rho}\right)\left(\frac{F\rho}{\mu^2}\right)^w$$

and Eq. (31) can be written

$$V = a_1 \left(\frac{\mu}{D\rho}\right)\left(\frac{F\rho}{\mu^2}\right)^{w_1} + a_2 \left(\frac{\mu}{D\rho}\right)\left(\frac{F\rho}{\mu^2}\right)^{w_2} + \cdots$$

or

$$\frac{DV\rho}{\mu} = a_1 \left(\frac{F\rho}{\mu^2}\right)^{w_1} + a_2 \left(\frac{F\rho}{\mu^2}\right)^{w_2} + \cdots$$

or

$$\frac{DV\rho}{\nu} = f\left(\frac{F\rho}{\mu^2}\right). \tag{32}$$

Thus, dimensional analysis indicates that the quantity $DV\rho/\mu$ depends only upon the quantity $F\rho/\mu^2$, therefore, if these two quantities are used as the coordinates, all the experimental data will fall upon a single curve.

Such a curve, determined by experiments with perhaps two or three different diameter spheres and two or three liquids, could be used to predict the maximum velocity V that a sphere of any other diameter submerged in any other liquid would attain if it were acted upon by any given force F. The quantity $F\rho/\mu^2$ would first have to be calculated from the given value of F and the values of ρ and μ for the liquid. The value of the quantity $DV\rho/\mu$ corresponding to this value of $F\rho/\mu^2$ could then be determined from the experimental curve, and, finally, the value of V could be calculated from this value of $DV\rho/\mu$.

TABLE 6.—NOMENCLATURE

c_p	Specific heat at constant pressure, B t u /(lb)(°F)
D	Inside or outside tube diameter, ft
d	Inside diameter of pipe, in
d_c	Diameter of coil, in
G	Weight velocity, lb./(hr)(sq ft of cross section)
g	Acceleration of gravity, 4 18 × 10⁸ ft per hr ²
h	Film coefficient of convection, B t u /(sq ft)(hr)(°F)
h_s	Scale coefficient, B t u /(sq ft)(hr)(°F)
k	Thermal conductivity, B t u /(ft)(hr)(°F)
N	Length of tube or height of surface, ft
n	Number of tubes directly over each other
P	Pressure, lb per sq ft abs
p	Pressure, atm abs
R	Gas constant, ft per °F
r	Latent heat of evaporation or condensation, B t u per lb
T	Temperature, °F abs
t	Temperature, °F.
t_f	Film temperature, °F
Δt	Temperature difference, °F
V	Linear velocity, ft per hr
v	Specific volume, cu ft per lb
W	Rate of condensation, lb /(sq ft)(hr)
W'	Rate of flow or rate of condensation, lb per hr per tube
β	Thermal coefficient of expansion, 1/°F
μ	Absolute viscosity, lb /(ft)(hr)
ν	Kinematic viscosity, sq ft per hr.
ρ	Density, lb per cu ft
σ	Surface tension, lb. per ft

31. Calculation of the Charts and Tables —The method of dimensional analysis illustrated in the preceding section can also be applied to the phenomenon of convection and has proved of great value in correlating the experimental data   Many investigators have expressed their results in the form of dimen-

sionless equations suggested by it, and certain of these equations have been used to calculate the charts and tables presented in this chapter. These equations together with the other data upon which the charts and tables are based are given in this section. The symbols used are defined in Table 6.

CASE 1. LIQUIDS HEATED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 22 and Tables 8 and 9 are based on the following equation recommended by McAdams:[1]

$$\frac{hD}{k} = 0.0225 \left(\frac{DV\rho}{\mu}\right)^{0.8} \left(\frac{c_p\mu}{k}\right)^{0.4}. \tag{33}$$

Each of the physical properties is evaluated at the average temperature of the liquid. This equation is valid for values of $DV\rho/\mu$ as low as 2,100, provided the value of $c_p\mu/k$ is less than 10. Otherwise, it is valid only for values of $DV\rho/\mu$ greater than 10,000. For the range of velocities given in Fig. 22, the liquids affected by these restrictions are listed in Table 10.

Although Eq. (33) is recommended only for horizontal tubes, the data of Stender[2] indicate that it can also be used to calculate conservative values for vertical tubes.

CASE 2. LIQUIDS COOLED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 23 and Tables 11, 12, and 13 are based on an equation exactly like Eq. (33) except with the exponent of the $c_p\mu/k$ term changed from 0.4 to 0.3. This equation was suggested by the work of Dittus and Boelter.[3]

CASE 3. GASES HEATED OR COOLED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 24 and Tables 14 and 15 are based on Eq. (33). This equation is valid for gases being either heated or cooled, provided the value of $G = V\rho$ is greater than $1,650p^{0.645}$. Values of

$$G_{\min} = (V\rho)_{\min} = 1,650p^{0.645}$$

are given in Table 16.

CASE 4. GASES AT ATMOSPHERIC PRESSURE HEATED OR COOLED INSIDE HORIZONTAL OR VERTICAL TUBES, TURBULENT FLOW.—Figure 25 and Tables 17 and 18 are also based on Eq.

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[1] McADAMS, W. H., "Heat Transmission," p. 169, McGraw-Hill Book Company, Inc., New York, 1933.

[2] *Ind. Eng. Chem.*, vol. 24, p. 1318, 1932.

[3] *Univ. Calif. Pub. Eng.*, vol. 2, p. 443, 1930.

(33), the equation being valid for gases if $V\rho$ is greater than $1,650\rho^{0.445}$ The gases are assumed to obey the perfect gas law
$$P_t = RT$$

CASE 5   LIQUIDS HEATED OR COOLED OUTSIDE SINGLE TUBES, DIRECTION OF FLOW NORMAL TO TUBE—Figure 26 and Tables 20 and 21 are based on the following equation
$$\frac{hD}{k} = 0.385\left(\frac{c_p\mu}{k}\right)^{0.3}\left(\frac{D V\rho}{\mu}\right)^{0.56}, \tag{34}$$

each of the physical properties being evaluated at the film temperature t_f The form of this equation was suggested by Ulsamer,[1] and the constant 0.385 was determined from Reiher's[2] equation for air
$$\frac{hD}{k} = 0.35\left(\frac{D V\rho}{\mu}\right)^{0.56}, \tag{35}$$

by taking $c_p\mu/k$ for air equal to 0.73 Equation (34) is recommended only for values of $D V\rho/\mu$ greater than 100 No data for liquids are available in this range but the data of Davis[3] on water and oil for the range $D V\rho/\mu$ from 0.1 to 100 agree very well with the data on air for the same range For the range of values of velocity, temperature, and diameter given in Fig 26 and Tables 20 and 21, $D V\rho/\mu$ always exceeds 100

CASE 6   GASES HEATED OR COOLED OUTSIDE SINGLE TUBES DIRECTION OF FLOW NORMAL TO TUBE—Figure 27 and Tables 22 and 23 are based on Eq (34) This equation is not recommended for values of $D V\rho/\mu$ less than 100, but for the range of values of velocity, temperature, and diameter given on the chart and in the tables, $D V\rho/\mu$ always exceeds this minimum value

CASE 7   GASES AT ATMOSPHERIC PRESSURE HEATED OR COOLED OUTSIDE SINGLE TUBES, DIRECTION OF FLOW NORMAL TO TUBE—Figure 28 and Tables 24 and 25 are based on Eq (34) and on the assumption that the gases obey the perfect gas law $Pv = RT$ For the range of values of velocity, temperature, and diameter given on the chart and in the tables, the product $D V\rho/\mu$ always exceeds the minimum permissible value of 100

CASE 8   LIQUIDS OR GASES HEATED OR COOLED OUTSIDE TUBE BUNDLES, DIRECTION OF FLOW NORMAL TO TUBES—The

[1] Forsch Gebiete Ing vol 3 p 94 1932
[2] Mitt Forschungsarb vol 269 p 20 1925
[3] Phil Mag vol 47 pp 972 and 1057 1924

data of Pierson, Huge, and Grimison[1] on air indicate that approximate values of the film coefficient h for fluids flowing normal to banks of tubes can be determined by multiplying the value of h for flow normal to single tubes by 1.3 if the tubes are staggered and by 1.2 if the tubes are in line.

The data of Reiher on air flowing normal to five rows of staggered tubes also show an increase in the film coefficient for tube bundles compared to the value for single tubes. Although his data for flow normal to five rows of tubes in line show a decrease compared to single tubes, Reiher found that the average value of the film coefficient for the entire tube bundle increased as the number of rows of tubes was increased. Consequently, had he used more than five rows of tubes, he might also have found the average coefficient for the entire bundle to be greater than for a single tube.

CASE 9. LIQUIDS OR GASES HEATED OR COOLED OUTSIDE TUBE BUNDLES, DIRECTION OF FLOW PARALLEL TO TUBES.—For flow parallel to the outside of tubes, Nusselt[2] recommends that film coefficients be determined from the data for flow inside tubes, using an equivalent inside diameter equal to four times the hydraulic radius. Equation (46) and Figs. (29) and (30) are based on this procedure.

CASE 10. LIQUIDS OR GASES HEATED OR COOLED IN ANNULAR SPACES, TURBULENT FLOW.—Film coefficients for fluids flowing through the annular space of a double-pipe heat exchanger, in which heat transfer is taking place between the fluid in the inner pipe and the fluid in the annular space, can be determined from the data for flow inside tubes, provided an equivalent diameter equal to four times the hydraulic radius is used for the inside diameter. Equation (47) is based on this procedure.

CASE 11. LIQUIDS HEATED OUTSIDE SINGLE HORIZONTAL TUBES, NATURAL CONVECTION.—Figure 31 and Tables 26 and 27 are based on the following equation recommended by McAdams:[3]

$$\frac{hD}{k} = 0.525 \left[\left(\frac{D^3 \rho^2 \beta g (\Delta t)}{\mu^2} \right) \left(\frac{c_p \mu}{k} \right) \right]^{0.25}, \tag{36}$$

[1] *Trans. A.S.M.E.*, vol. 59, pp. 563, 573, and 583, 1937.

[2] *Z. Ver. deut. Ing.*, vol. 57, p. 199, 1913, and vol. 61, p. 685, 1917.

[3] McADAMS, *op. cit.*, p. 249.

each of the physical properties being evaluated at the film temperature t_f This equation is valid only for values of

$$\left[\left(\frac{D^3\rho^2\beta g(\Delta t)}{\mu^2}\right)\left(\frac{c_p\mu}{k}\right)\right]$$

greater than 1,000 This value is exceeded for all the liquids listed for the ranges of temperature difference, film temperature, and diameter given on the chart and in the tables

CASE 12   GASES HEATED OUTSIDE SINGLE HORIZONTAL TUBES, NATURAL CONVECTION—Figure 32 and Tables 28, 29, and 30 are based on Eq (36) and on the assumption that the gases obey the perfect gas law $Pv = RT$ For the ranges of temperature difference, film temperature, diameter, and pressure given, none of the gases listed falls below the lower limit for which Eq (36) is valid

CASE 13   LIQUIDS HEATED INSIDE OR OUTSIDE VERTICAL TUBES OR ON VERTICAL PLATES LOW VELOCITIES OR NATURAL CONVECTION ONLY—Figure 33 and Table 31 are based on the following equation of Colburn and Hougen [1]

$$h = 0\ 128\left(\frac{k^3\rho^2 c_p\beta g(\Delta t)}{\mu}\right)^{\frac{1}{4}}\tag{37}$$

Each of the physical properties is evaluated at the film temperature t_f This equation may be used either for natural convection or for forced convection, provided the velocities are low The equation is based on data for upward flow inside tubes but it may also be used to predict conservative values for downward flow It may also be used for predicting film coefficients for fluids outside tubes since it agrees very well with the equation of Rice [2] for air heated by natural convection outside pipes

CASE 14   GASES HEATED INSIDE OR OUTSIDE VERTICAL TUBES OR ON VERTICAL PLATES, NATURAL CONVECTION—Figure 34 and Tables 32 and 33 are based on Eq (36) and on the assumption that the gases obey the perfect-gas law $Pv = RT$

CASE 15   LIQUIDS HEATED OR COOLED INSIDE COILS, TURBULENT FLOW—For any given velocity, the flow inside cooled tubes is considerably more turbulent than that inside straight tubes This results in an increase in the value of the film coefficient h,

[1] Ind Eng Chem vol 22 p 522 1930
[2] Trans Am Inst Elec Eng, vol 43 p 131 1924

the exact amount depending upon the radius of the coil. Richter[1] found that over-all coefficients for heat transfer from water to water inside a helical double-pipe exchanger were about 20 per cent larger than over-all coefficients for straight double-pipe exchangers. This suggests that approximate values for the film coefficient h can be obtained by multiplying the value of h for straight tubes by a factor such as 1.2.

CASE 16. GASES HEATED OR COOLED INSIDE COILS, TURBULENT FLOW.—Based on the work of Jeschke, McAdams[2] suggests that values of the film coefficient h for the turbulent flow of gases inside coils be obtained by multiplying the value of h for straight tubes by the factor

$$\left[1 + 3.54\left(\frac{d}{d_c}\right)\right],$$

where d/d_c is the ratio of the diameters of the pipe and the coil.

CASE 17. LIQUIDS OR GASES HEATED OR COOLED OUTSIDE COILS, NATURAL OR FORCED CONVECTION.—The data of Gibson[3] on air flowing normal to coils agree very well with Eq. (35). This suggests that film coefficients for fluids flowing outside coils may be assumed approximately equal to the film coefficients for fluids flowing normal to single tubes.

CASE 18. AIR HEATED ON HORIZONTAL PLATES, NATURAL CONVECTION.—Heating air by natural convection on large flat plates, Griffiths and Davis[4] found that the coefficient h for horizontal plates facing *upward* was about 27 per cent higher than for vertical plates; and for horizontal plates facing *downward*, about 33 per cent lower.

CASE 19. PETROLEUM OILS HEATED INSIDE HORIZONTAL OR VERTICAL TUBES, STREAMLINE FLOW.—Figure 35 and Tables 34 and 35 are based on the following empirical equation suggested by McAdams:[5]

$$\frac{hD}{k} = 6.2\left(\frac{W'c_p}{kN}\right)^{0.2}, \tag{38}$$

[1] *Am. Inst. Chem. Eng.*, vol. 12, Part II, p. 147, 1919.

[2] McADAMS, *op. cit.*, p. 179.

[3] *Phil. Mag.*, Ser. 6, vol. 47, p. 324, 1924.

[4] Food Investigation Board, *Dept. Sci. Ind. Research, Special Rept.* 9, His Majesty's Stationery Office, London, 1922.

[5] McADAMS, *op. cit.*, p. 210.

in which the coefficient h is based on the arithmetic mean of the terminal temperature differences, and c_p and k are evaluated at the average temperature of the oil after mixing   This equation is valid only if $W'c_p/k\lambda$ exceeds 30 and if $DV\rho/\mu$ is less than 2,100   For the range of gravities and W'/λ given in Fig 35, the product $W'c_p/k\lambda$ always exceeds 30   The values of W'/D for which $DV\rho/\mu$ equals 2 100 are given in Table 36

Case 20   Petroleum Oils Cooled inside Horizontal or Vertical Tubes Streamline Flow —The data plotted by McAdams[1] on cooling hydrocarbon oils flowing at low velocities inside either horizontal or vertical tubes show that approximate values of the film coefficient h for cooling can be obtained by multiplying the values of h for heating calculated by Eq (38), by 0 7

Case 21   Condensation of Pure Saturated Vapors on Horizontal Tubes —Figure 36 and Tables 37, 38, and 39 are based on theoretical equation of Nusselt[2] for filmwise condensation of pure saturated vapors on horizontal tubes

$$h = 0 725 \left[\frac{r\rho^2 k^3 g}{n\mu D(\Delta t)}\right]^{\frac{1}{4}}$$ (39)

and on the equation

$$h(\Delta t) = W r$$ (40)

The latent heat of condensation r is evaluated at the temperature of the vapor but the other physical properties are evaluated at the temperature of the condensate film   Equation (39) gives conservative values the deviation from test data increasing with the rate of condensation

Case 22   Condensation of Pure Saturated Vapors on Vertical Tubes or Plates —Figure 37 and Tables 41 and 42 are based on the following equation given by Kirkbride[3] for filmwise condensation of pure vapors on vertical tubes

$$h\left(\frac{\mu^2}{k^3\rho^2 g}\right)^{\frac{1}{4}} = 0 0084 \left(\frac{W}{\mu D}\right)^{0 4}$$ (41)

Each of the physical properties is evaluated at the temperature of the condensate film   Figure 36 and Tables 37 and 43 are

[1] McAdams op cit p 210
[2] Z Ver deut Ing vol 60 pp 541 and 569 1916
[3] Trans Am Inst Chem Eng vol 32 p 170 1936

based on the theoretical equation of Nusselt for filmwise condensation of pure saturated vapors on vertical surfaces

$$h = 0.943 \left[\frac{r\rho^2 k^3 g}{N\mu(\Delta t)} \right]^{\frac{1}{4}} \qquad (42)$$

and on Eq. (40). The latent heat of condensation is evaluated at the temperature of the vapor, and the other physical properties are evaluated at the temperature of the condensate film. Equations (41) and (42) both yield the same value of the film coefficient h when $W'/\mu D$ is equal to 1,020. Equation (41) is recommended if $W'/\mu D$ is greater than 1,020, and Eq. (42) is recommended if $W'/\mu D$ is smaller. The values of WN for which $W'/\mu D$ is equal to 1,020 are given in Table 40.

CASE 23. LIQUIDS BOILING ON HORIZONTAL OR VERTICAL PLATES.—Figure 38 and Table 44 are based on the equation of Jakob[1] for evaporation on flat plates free from scale, occurring after the equipment has been in operation for some time:

$$\frac{h}{k} \left(\frac{\sigma}{\rho' - \rho''} \right)^{0.5} = 31.6 \frac{\nu_s}{\nu} \left[\left(\frac{\sigma}{\sigma_a} \right) \left(\frac{\rho_a' - \rho_a''}{\rho' - \rho''} \right) \frac{h(\Delta t)}{\rho_a'' r_a w_a} \right]^{0.8}, \quad (43)$$

where the prime denotes saturated liquid; the double prime, saturated vapor; the subscript a, atmospheric pressure; the subscript s, saturated water at 212°F.; and w_a is equal to 918 ft. per hr. and represents the product of the mean frequency of formation of bubbles at any point on the heating surface times the mean diameter of the vapor bubbles leaving the surface. This equation gives quite conservative values of the film coefficient.

CASE 24. LIQUIDS BOILING INSIDE TUBES.—According to Jakob, values of the coefficient h for evaporation in tubes are about 25 per cent higher than for evaporation on flat plates.

CASE 25. SCALE COEFFICIENTS.—McAdams[2] suggests that the scale coefficients given be used if specific information is not available.

32. Use of the Charts and Tables.—Values of the film coefficient h for various fluids under a number of different conditions can be readily determined by means of the charts and tables that follow. Table 7 will be found useful as an index to the various cases covered.

[1] *Tech. Bull. Armour Inst. Tech.*, vol. 2, no. 1, 1939.
[2] MCADAMS, *op. cit.*, p. 151.

TABLE 7.—OUTLINE OF THE KINDS OF CONVECTION FOR WHICH FILM COEFFICIENTS ARE GIVEN

a Convection involving no change of phase

Kind of fluid	Kind of surface	Position of surface face horizontal or vertical	Kind of fluid motion	Fluid heated or cooled	Case No	Page
Liquids	Inside tubes	Horizontal	Turbulent	Heated Cooled	1 2	80 82
			Streamline (pet ods only)	Heated Cooled	19 20	105 105
		Vertical	Turbulent	Heated Cooled	1 2	80 82
			Streamline (pet ods only)	Heated Cooled	19 20	105 105
			Natural conv	Heated	13	100
	Outside single tubes	Horizontal	Normal to tube Natural conv	Either Heated	5 11	88 96
		Vertical	Normal to tube Natural conv	Either Heated	5 13	88 100
	Outside tube bundles	Horizontal	Normal to tube Parallel to tubes	Either Either	8 9	94 94
		Vertical	Normal to tubes Parallel to tubes Natural conv	Either Either Heated	8 9 13	94 94 100
	Annular spaces	Either	Turbulent	Either	10	94
	Inside coils	Either	Turbulent	Either	15	104
	Outside coils	Either	Forced conv Natural conv	Either Heated	17 17	104 104
	Flat plates	Vertical	Natural conv	Heated	13	100
Gases	Inside tubes	Horizontal	Turbulent	Either	3 or 4	84
		Vertical	Turbulent Natural conv	Either Heated	3 or 4 14	84 102
	Outside single tubes	Horizontal	Normal to tube Natural conv	Either Heated	6 or 7 12	90 98
		Vertical	Normal to tube Natural conv	Either Heated	6 or 7 14	90 102
	Outside tube bundles	Horizontal	Normal to tube Parallel to tubes	Either Either	8 9	94 94
		Vertical	Normal to tubes Parallel to tubes Natural conv	Either Either Heated	8 9 14	94 94 102
	Annular spaces	Either	Turbulent	Either	10	94
	Inside coils	Either	Turbulent	Either	16	104
	Outside coils	Either	Forced conv Natural conv	Either Heated	17 17	104 104
	Flat plates	Horizontal (air only)	Natural conv	Heated	18	104
		Vertical	Natural conv	Heated	14	102

b Condensation of saturated vapors

Kind of surface	Position of surface, horizontal or vertical	Case No	Page
Inside tubes	Vertical	22	108
Outside tubes	{ Horizontal	21	106
	{ Vertical	22	108
Flat plates	Vertical	22	108

c Evaporation of liquids

Kind of surface	Position of surface, horizontal or vertical	Case No	Page
Inside tubes	Vertical	24	110
Flat plates	Either	23	110

d Scale coefficients, Case 25, page 112

The same method of presentation is used throughout: Values of the film coefficient h_0 at some arbitrarily chosen "base" condition are given on the charts, and correction factors for changing these values to conditions other than the base conditions are given in the tables accompanying the charts. For example, in Case 1 the film coefficient h for any of the liquids listed, flowing at any velocity V and temperature t inside a tube of any inside diameter d, can be found from the equation

$$h = h_0 \times F_t \times F_d, \tag{44}$$

where h = the desired film coefficient, B.t.u./(sq. ft.)(hr.)(°F.).

h_0 = the film coefficient for the given liquid at the "base" conditions and at the velocity V, read from Fig. 22.

F_t = the temperature correction factor corresponding to the given temperature t, read from Table 8.

F_d = the diameter correction factor corresponding to the given inside tube diameter d, read from Table 9.

Many of the charts and tables involve either the temperature difference Δt between the surface of the retaining wall and the main body of the fluid or the *film temperature* t_f, which is arbitrarily defined as the arithmetic average of the temperature of the surface of the retaining wall and the temperature of the main body of the fluid. In order to evaluate either of these quantities, the temperature of the surface of the retaining wall must be known. Sufficiently accurate results can frequently be obtained by assuming some value that appears reasonable for this temperature. If more accurate results are desired, the

exact temperature of the retaining wall can be calculated by trial and error as follows Neglecting the temperature drop through the retaining wall itself and using the nomenclature shown in Fig 21, (1) a tentative value is assumed for the temperature t_w of the retaining wall (2) Based on this value, the values of Δt_1, Δt_2 and t_f are calculated (3) The film coefficients h_1 and h_2 corresponding to these values are determined from the
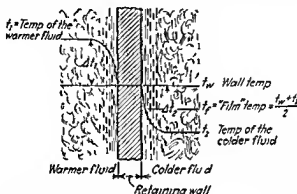
Fig 21 —Temperature gradient through two fluids separated by a retaining wall

charts and tables (4) Using these values of h_1 and h_2, a more accurate value of t_w is calculated by the equation

$$t_w = \frac{h_1 t_1 + h_2 t_2}{h_1 + h_2} \tag{45}$$

This value of t_w will usually be sufficiently accurate for practical purposes Using it values of Δt_1, Δt_2 and t_f can again be calculated, and the final value of h_1 and h_2 can then be determined

Equation (45) is based on the simplifying assumption that the temperature drop through the retaining wall is negligible and that the areas of the two surfaces of the wall are approximately equal By Eq (23) in Chap III,

$$q = h_1 A (t_1 - t_w) = h_2 A (t_w - t_2)$$

Equation (45) is obtained by solving the last equation for t_w.

Illustrative Problem 1 (Shell and tube Heat Exchanger) —Design a shell and tube heat exchanger to cool 40 gal of water per minute from 55°F by means of 60 gal per min of calcium chloride brine (25 per cent solution) initially at 30°F The tubes are to be $\frac{3}{4}$ in o d with No 16 B W G walls and are to be made of brass

Data.—Specific heat of water = 1.00 B.t.u./(lb.)(°F.); specific heat of the brine = 0.68 B.t.u./(lb.)(°F.); density of water = 8.35 lb. per gal.; density of the brine = 10.3 lb. per gal. Conversion factor: 1 cu. ft. = 7.48 gal.

Solution.—The required rate of heat transfer can be calculated from the rate of flow, specific heat, and temperature drop of the water. Thus,

$$q = (40 \times 8.35 \times 60) \times 1.00 \times (55 - 45)$$
$$= 200,000 \text{ B.t.u. per hr.}$$

Since this is the rate at which the brine will absorb heat, the final temperature of the brine can be calculated by the equation

$$200,000 = (60 \times 10.3 \times 60) \times 0.68 \times (t - 30),$$

or

$$t = 38°F.$$

Let the brine flow through the tubes and the water flow through the shell. The velocity of the brine should be at least 3 ft. per sec. in order to assure turbulent flow (see Table 10). This velocity will be obtained if 19 tubes are used per pass. Thus, since the internal cross-sectional area of each tube is 0.302 sq. in.,

$$\frac{60}{7.48 \times 60} = \frac{19 \times 0.302}{144} \times V,$$

or

$$V = 3.4 \text{ ft. per sec.}$$

First Design.—Let each liquid make a single pass through the exchanger. If the tubes are spaced at the apexes of 1-in. equilateral triangles, the shell may be made of 5-in. pipe. Since the internal cross-sectional area of 5-in. pipe is 20.0 sq. in. and the external cross-sectional area of each tube is 0.442 sq. in., the velocity of the water can be calculated by the equation

$$\frac{40}{7.48 \times 60} = \frac{(20.0 - 19 \times 0.442)}{144} \times V,$$

or

$$V = 1.1 \text{ ft. per sec.}$$

The film coefficient h_1 for the brine can be determined by Case 1. Thus, from Fig. 22, $h_0 = 680$ B.t.u./(sq. ft.)(hr.)(°F.); from Table 8, $F_t = 0.68$ (average temperature of the brine = 34°F.); and from Table 9, $F_d = 1.10$ (inside tube diameter = 0.620 in.). Hence,

$$h_1 = 680 \times 0.68 \times 1.10$$
$$= 510 \text{ B.t.u./(sq. ft.)(hr.)(°F.)}.$$

The film coefficient h_2 for the water can be determined by Cases 9 and 2. Thus, referring to Fig. 29, $x = 1$ in., $d = 0.75$ in., $x/d = 1.33$, and therefore $d_e = 0.72$ in. Then, from Fig. 23, $h_0 = 290$ B.t.u./(sq. ft.)(hr.)(°F.); from Table 11, $F_t = 0.66$ (average temperature of the water = 50°F.); and from Table 12, $F_d = 1.08$ (equivalent inside diameter = 0.72 in.). Hence,

$$h_2 = 290 \times 0.66 \times 1.08$$
$$= 207 \text{ B.t.u./(sq. ft.)(hr.)(°F.)}.$$

The mean temperature difference between the steam and the water can be determined from Fig 17 [or calculated by Eq (30)] Thus,

$$\Delta t = 227 - 60 = 167°F ,$$

$\Delta t' = 227 - 180 = 47°F$, and $\Delta t_m = 9.5°F$ The *mean* temperature of the water may be taken, therefore, as $227 - 95 = 132°F$

In order to limit the pressure drop, the velocity of the water in the tubes should not exceed 2 5 ft per sec This requirement is satisfied if 28 tubes are used per pass Thus, since the cross-sectional area of each tube is 0 334 sq in ,

$$\frac{4\,000}{7\,48 \times 3\,600} = \frac{28 \times 0\,334}{144} \times V,$$

or

$$1 = 2\,3 \text{ ft per sec}$$

In order to limit the length of the tubes, let the heater be two pass

The film coefficient h_1 for the water can be determined by Case 1 From Fig 22, $h_1 = 610$ B t u /(sq ft)(hr)(°F), from Table 8

$$F_1 = 1\,14 \text{ (at } 132°F),$$

and from Table 9 $F_4 = 1\,09$ (inside tube diameter = 0 652 in) Hence,

$$h_1 = 610 \times 1\,14 \times 1\,09$$
$$= 760 \text{ B t u /(sq ft)(hr)(°F)}$$

The film coefficient h_2 for the steam can be determined by Case 21 Since this coefficient depends upon the tube-wall temperature t_e, a trial and-error solution is necessary

As a first trial assume $t_e = 200°F$ The inside tube surface A_1 corresponding to this value of t_e can be calculated from the film coefficient h_1 by Eq (23)

$$4\,000\,000 = 760 \times A_1 \times (200 - 132),$$

or

$$A_1 = 77\,5 \text{ sq ft}$$

The corresponding outside tube surface is

$$77\,5 \times \left(\frac{0\,750}{0\,652}\right) = 89\,2 \text{ sq ft}$$

and the rate of condensation of the steam

$$W = \frac{4\,170}{89\,2} = 47 \text{ lb /(sq ft)(hr)}$$

Then from Fig 36 $h_2 = 1450$ B t u /(sq ft)(hr)(°F), from Table 37 $F_1 = 1\,47$ (at $t_e = 200°F$), from Table 38 $F_4 = 1\,10$, and from Table 39 $F_e = 0\,55$ (assuming $n = 6$) Hence,

$$h_2 = 1450 \times 1\,47 \times 1\,10 \times 0\,55$$
$$= 1290 \text{ B t u /(sq ft)(hr)(°F)}$$

The tube wall temperature can now be calculated by Eq. (45):

$$t_w = \frac{760 \times 132 + 1290 \times 227}{760 + 1290}$$
$$= 192°F.$$

Since this does not agree with the value initially assumed, the calculations must be repeated.

As a second trial, assume $t_w = 192°F$. By Eq. (23),

$$4,000,000 = 760 \times A_1 \times (192 - 132),$$

or

$$A_1 = 87.7 \text{ sq. ft.}$$

The corresponding outside surface $A_2 = 87.7 \times (0.750/0.652) = 101$ sq. ft., and the rate of condensation

$$W = \frac{4,170}{101} = 41 \text{ lb./(sq. ft.)(hr.)}.$$

From Fig. 36, $h_0 = 1500$ B.t.u./(sq. ft.)(hr.)(°F.); and from Table 37,

$$F_t = 1.43 \text{ (at } t_w = 192°F.).$$

Hence,

$$h_2 = 1500 \times 1.43 \times 1.10 \times 0.55$$
$$= 1300 \text{ B.t.u./(sq. ft.)(hr.)(°F.)},$$

and by Eq. (45),

$$t_w = \frac{760 \times 132 + 1300 \times 227}{760 + 1300}$$
$$= 192°F.$$

Since this agrees with the value initially assumed, the preceding calculations are correct, and the outside tube surface required is 101 sq. ft. Since 5.093 lin. ft. of ¾-in. tubing have 1 sq. ft. of outside surface, the total length of tubing required is $5.093 \times 101 = 513$ ft., and the length of each tube must be $513/_{56} = 9.2$ ft. (Note that in the foregoing solution the temperature drop through the tube wall has been neglected.)

Illustrative Problem 3 (Oil Heater).—Design a shell-and-tube type heater to heat 800 gal. per hr. of fuel oil having a gravity of 30° A.P.I. and a viscosity of 150 Saybolt Universal seconds at 100°F. from 50 to 200°F. by means of steam at 10 lb. per sq. in. gauge pressure (temperature = 242°F.). The tubes are to be ¾ in. o.d. with No. 16 B.W.G. walls, and the oil is to flow through the tubes.

Data.—Mean specific heat of the oil = 0.47 B.t.u./(lb.)(°F.), and density of the oil = 7.30 lb. per gal.

Solution.—The required rate of heat transfer can be calculated from the rate of flow, specific heat, and temperature rise of the oil. Thus,

$$q = (800 \times 7.30) \times 0.47 \times (200 - 50)$$
$$= 410,000 \text{ B.t.u. per hr.}$$

The length of the heater will depend upon the number of tubes used Tentatively, let the heater consist of 90 tubes inclosed in a 12-in shell, and let the oil make two passes through the tubes

Since the flow of the oil will be streamline (see Table 36), the film coefficient h_1 for the oil can be determined by Case 19   Since the value of h_3 depends upon the length of the tubes, a trial and error solution is necessary As a first trial, assume that the tubes will be 10 ft long   Then, remembering that the oil will make two passes through the tubes,

$$W' = \frac{(800 \times 7\ 30)}{45} = 130 \text{ lb per hr per tube,}$$

$N = 2 \times 10 = 20$ ft , $W'/N = \frac{130}{20} = 6\ 5$ lb /(hr)(tube)(ft), and from Fig 35 $h_1 = 12\ 0$ B t u /(sq ft)(hr)(°F)   From Table 34, $F_4 = 1\ 00$, and from Table 35, $F_d = 1\ 62$ (inside tube diameter $= 0\ 620$ in)   Hence,

$$h_1 = 12\ 0 \times 1\ 00 \times 1\ 62$$
$$= 19\ 5 \text{ B t u /(sq ft)(hr)(°F)}$$

The film coefficient h_2 for the condensing steam can be determined by Case 21   However, since it will be quite large compared with h_1 h_2 will have very little effect on the over all coefficient U [i e the $1/h_2$ term in Eq (27) will be almost negligible], so its exact value need not be determined   Instead, assume $h_2 = 3000$ B t u /(sq ft)(hr)(°F)

The over all coefficient U, based on the outside tube surface, can be calculated by Eq (27)   Neglecting the L/k term,

$$\frac{1}{U} = \frac{0\ 750}{19\ 5 \times 0\ 620} + \frac{1}{3000},$$

or

$$U = 16\ 1 \text{ B t u /(sq ft)(hr)(°F)}$$

Since the film coefficient h_1 for the oil is based on the *arithmetic* mean of the terminal temperature differences the mean temperature difference between the oil and the steam may be calculated as follows

$$\Delta t = 242 - 50 = 192°F,$$

$\Delta t = 242 - 200 = 42°F$ and $\Delta t_{mean} = (192 + 42) 2 = 117°F$
The outside tube surface required can now be calculated by Eq (24)

$$410\ 000 = 16\ 1 \times A \times 117,$$

or

$$A = 218 \text{ sq ft}$$

Since 5 093 lin ft of $^3\!/_4$ in tubing is required per square foot of outside surface the total length of tubing required is 5 093 × 218 = 1 110 ft , and the length of each tube must be , 1,110/90 = 12 3 ft   Although this value does not agree with the value assumed in determining h_1 it is not necessary to repeat the calculations because changing the tube length from 10 to 12 3 ft would have an almost negligible effect on h_1 (see Fig 35)   Since the tubes are of a reasonable length, this design would be satisfactory

Illustrative Problem 4 (Double-pipe Heat Exchanger).—A double-pipe heat exchanger is to be used to cool 10 gal. of water per minute from 55 to 45°F. by means of 10 gal. per min. of calcium chloride brine (25 per cent solution) initially at 10°F. How long must the exchanger be if it is made of 1¼- and 2½-in. wrought-iron pipe?

Data.—Specific heat of water = 1.00 B.t.u./(lb.)(°F.); specific heat of the brine = 0.68 B.t.u./(lb.)(°F.); density of water = 8.35 lb. per gal.; density of the brine = 10.3 lb. per gal. Conversion factor:

$$1 \text{ cu. ft.} = 7.48 \text{ gal.}$$

Solution.—The required rate of heat transfer can be calculated from the rate of flow, specific heat, and temperature drop of the water:

$$q = (10 \times 8.35 \times 60) \times 1.00 \times (55 - 45)$$
$$= 50,000 \text{ B.t.u. per hr.}$$

Since this is the rate at which the brine will absorb heat, the final temperature of the brine can be calculated by the equation

$$50,000 = (10 \times 10.3 \times 60) \times 0.68 \times (t - 10),$$

or

$$t = 22°F.$$

Let the brine flow through the inner pipe, and the water through the annular space. The velocity of the brine can be calculated by the equation (internal cross-sectional area of a 1¼-in. pipe = 1.495 sq. in.)

$$\frac{10}{7.48 \times 60} = \frac{1.495}{144} \times V,$$

or

$$V = 2.2 \text{ ft. per sec.}$$

Similarly, the velocity of the water can be calculated by the equation (internal cross-sectional area of a 2½-in. pipe = 4.788 sq. in.; external cross-sectional area of a 1¼-in. pipe = 2.164 sq. in.)

$$\frac{10}{7.48 \times 60} = \frac{(4.788 - 2.164)}{144} \times V,$$

or

$$V = 1.2 \text{ ft. per sec.}$$

The film coefficient h_1 for the brine can be determined by Case 1. Thus, from Fig. 22, $h_0 = 470$ B.t.u./(sq. ft.)(hr.)(°F.); from Table 8,

$$F_t = 0.60 \text{ (average brine temperature } = 16°F.);$$

and from Table 9, $F_d = 0.94$ (inside pipe diameter = 1.38 in.). Hence,

$$h_1 = 470 \times 0.60 \times 0.94$$
$$= 265 \text{ B.t.u./(sq. ft.)(hr.)(°F.).}$$

The film coefficient h_2 for the water can be determined by Cases 10 and 2. Thus, by Eq. (47) (inside diameter of a 2½-in. pipe = 2.469 in.; outside

diameter of a $1\frac{1}{4}$ in pipe = 1 660 in,

$$d_e = \frac{(2\,469)^2 - (1\,660)^2}{1\,660}$$

$$= 2\,02\text{ in}$$

From Fig 23, $h_4 = 310$ B t u /(sq ft)(hr)(°F), from Table 11,

$$F_z = 0\,66\text{ (average water temperature} = 50°F\text{),}$$

and from Table 12, $F_d = 0\,87$ (for $d_e = 2\,02$ in) Hence,

$$h_2 = 310 \times 0\,66 \times 0\,87$$
$$= 178\text{ B t u /(sq ft)(hr)(°F)}$$

The over all coefficient U based on the outside surface of the inner pipe can be calculated by Eq (27) Since the pipe wall is 0 140 in thick, and since the thermal conductivity of wrought iron is about 35 B t u /(ft)(hr) (°F),

$$\frac{1}{U} = \frac{1\,660}{265 \times 1\,380} + \frac{0\,140}{12 \times 35} + \frac{1}{178},$$

or

$$U = 95\text{ B t u /(sq ft)(hr)(°F)}$$

For counterflow, the terminal temperature differences are

$$\Delta t' = 45 - 10 = 35°F$$

and $\Delta t'' = 55 - 22 = 33°F$, so the mean temperature difference = 34°F

The pipe surface required can now be calculated by Eq (24)

$$50\,000 = 95 \times A \times 34$$

or

$$A = 15\,5\text{ sq ft}$$

Since 2 301 lin ft of $1\frac{1}{4}$ in pipe have 1 sq ft of outside surface, the total length of the heat exchanger must be 2 301 × 15 5 = 35 6 ft The exchanger could consist, for example, of four sections each 9 ft long

Illustrative Problem 5 (Pipe Coil) —One hundred cubic feet of free air per minute, compressed to a pressure of 50 lb per sq in gauge is to be cooled from 210 to 90°F in a coil submerged in water at 80°F How long must the coil be if it is made of $1\frac{1}{4}$ in steel pipe bent into a 2-ft diameter helix?

Data —Specific heat of air at constant pressure = 0 24 B t u /(lb)(°F), density of the free air = 0 075 lb per cu ft

Solution —The weight of air flowing through the coil will be

$$100 \times 60 \times 0\,075 = 450\text{ lb per hr}$$

The required rate of heat transfer can be calculated from this value, the specific heat, and the temperature drop of the air

$$q = 450 \times 0\,24 \times (210 - 90)$$
$$= 13\,000\text{ B t u per hr}$$

The film coefficient h_1 for the air can be determined by Cases 16 and 3 Since the internal cross-sectional area of $1\frac{1}{4}$ in pipe is 1 495 sq in , the

weight velocity of the air

$$G = \frac{450 \times 144}{3,600 \times 1.495} = 12.0 \text{ lb./(sec.)(sq. ft.)}$$

(Note that this value is larger than the minimum permissible value given in Table 16.) Then from Fig. 24, $h_0 = 28$ B.t.u./(sq. ft.)(hr.)(°F.); from Table 14, $F_t = 1.02$ (average temperature of the air = 150°F.); and from Table 15, $F_d = 0.94$ (inside pipe diameter = 1.38 in.). Hence, for the *coil*,

$$h_1 = \left[1 + 3.54 \left(\frac{1.38}{24} \right) \right] \times 28 \times 1.02 \times 0.94$$
$$= 32 \text{ B.t.u./(sq. ft.)(hr.)(°F.).}$$

The film coefficient h_2 for the water can be determined by Cases 17 and 11. The average pipe wall temperature t_w must first be determined by the method explained in Sec. 32: Assume $t_w = 90°$F. Then, from Fig. 31, $h_0 = 85$ B.t.u./(sq. ft.)(hr.)(°F.) [$\Delta t = 90 - 80 = 10°$F.]; from Table 26, $F_t = 0.93$ [average film temperature = $(90 + 80)/2 = 85°$F.]; and from Table 27, $F_d = 0.88$ (outside pipe diameter = 1.66 in.). Hence, as a first approximation,

$$h_2 = 85 \times 0.93 \times 0.88$$
$$= 70 \text{ B.t.u./(sq. ft.)(hr.)(°F.).}$$

A more accurate value of t_w can now be determined by Eq. (45):

$$t_w = \frac{32 \times 150 + 70 \times 80}{32 + 70}$$
$$= 102°\text{F.}$$

Redetermining h_2 for this value of t_w, from Fig. 31,

$$h_0 = 105 \text{ B.t.u./(sq. ft.)(hr.)(°F.)}$$

[$\Delta t = 102 - 80 = 22°$F.]; and from Table 26,

$$F_t = 0.96 \text{ [average film temperature} = \frac{(102 + 80)}{2} = 91°\text{F.].}$$

Finally,

$$h_2 = 105 \times 0.96 \times 0.88$$
$$= 89 \text{ B.t.u./(sq. ft.)(hr.)(°F.).}$$

The over-all coefficient U, based on the outside pipe surface, can be calculated by Eq. (27): Thus, since the pipe wall is 0.140 in. thick, and since the thermal conductivity of steel is about 26 B.t.u./(ft.)(hr.)(°F.),

$$\frac{1}{U} = \frac{1.66}{32 \times 1.38} + \frac{0.140}{12 \times 26} + \frac{1}{89},$$

or

$$U = 20 \text{ B.t.u./(sq. ft.)(hr.)(°F.) (clean pipe).}$$

Allowance should be made for the presence of scale on the outside surface of the pipe. The scale coefficient h_s can be determined by Case 25: Assum-

ing that a deposit of scale 0 1 in thick will gradually accumulate,

$$h_s = \frac{1 \times 12}{0 1} = 120 \text{ B t u /(sq ft)(hr)(°F)}$$

Then, by Eq (29),

$$\frac{1}{U_s} = \frac{1}{20} + \frac{1}{120},$$

or

$$U_s = 17 \text{ B t u /(sq ft)(hr)(°F) (scaly pipe)}$$

The mean temperature difference between the air and the water can be determined from Fig 17 (or calculated by Eq 30) Thus,

$$\Delta t = 210 - 80 = 130°F ,$$

$$\Delta t'' = 90 - 80 = 10°F , \quad \text{and} \quad \Delta t_m = 47°F$$

The outside pipe surface required can now be calculated by Eq (24)

$$13 000 = 17 \times A \times 47,$$

or

$$A = 16 3 \text{ sq ft}$$

Since 2 301 lin ft of 1½ in pipe have 1 sq ft of outside surface, the total length of pipe needed in the coil is 2 301 × 16 3 = 37 5 ft

Illustrative Problem 6 (Evaporator)—A weak salt solution is to be concentrated in a vertical short tube evaporator (see Sec 405) The evaporator will operate at a pressure of 24 in Hg absolute and will be supplied with saturated steam at 10 lb per sq in gauge Assuming that the solution has approximately the same properties as water determine the number of 2-in o d × No 14 B W G brass tubes 5 ft long required to evaporate 4,000 lb of water per hour

Data—Steam temperature = 239°F , solution temperature = 201°F , heat of condensation of the steam = 952 B t u per lb , heat supplied to the solution per lb of water evaporated = 1100 B t u

Solution—The required rate of heat transfer

$$q = 4 000 \times 1 100 = 4,400,000 \text{ B t u per hr },$$

and the weight of steam required is 4 400,000/952 = 4,620 lb per hr

The film coefficient h_i for the solution (i e , for water) can be determined by Case 24, and the film coefficient h_s for the steam can be determined by Case 22 Since both coefficients depend upon the tube-wall temperature t_w, a trial and error solution is necessary

As a first trial, assume $t_w = 225°F$ From Fig 38,

$$h_i = 2200 \text{ B t u /(sq ft)(hr)(°F)} \quad (\Delta t = 225 - 201 = 24°F),$$

and from Table 44, $F_p = 0 94$ (pressure = $^{24}/_{50} = 0 8 \text{ atm abs }$) Hence

$$h_i = 1 25 \times 2200 \times 0 94$$
$$= 2600 \text{ B t u /(sq ft)(hr)(°F)}$$

The inside tube surface A_1 corresponding to this value of h_1 can be calculated by Eq. (23):

$$4,400,000 = 2,600 \times A_1 \times (225 - 201),$$

or

$$A_1 = 70.5 \text{ sq. ft.}$$

The corresponding outside tube surface (inside tube diameter $= 1.834$ in.) is $70.5 \times (2.000/1.834) = 77.0$ sq. ft., and the rate of condensation of the steam $W = 4,620/77.0 = 60.0$ lb./(sq. ft.)(hr.). Referring to Table 40, since $W \times N = 60.0 \times 5 = 300$, method a of Case 22 must be used to determine h_2. From Fig. 37, $h_0 = 420$ B.t.u./(sq. ft.)(hr.)(°F.); from Table 41, $F_t = 3.19$ (at $t_w = 225$°F.); and from Table 42, $F_n' = 0.75$. Hence, for clean tubes, $h_2 = 420 \times 3.19 \times 0.75 = 1000$ B.t.u./(sq. ft.)(hr.)(°F.). Combining this value with a scale coefficient

$$h_s = 3000 \text{ B.t.u./(sq. ft.)(hr.)(°F.) [from Case 25],}$$
$$\frac{1}{h_2'} = \frac{1}{1000} + \frac{1}{3000},$$

or

$$h_2' = 750 \text{ B.t.u./(sq. ft.)(hr.)(°F.).}$$

The tube-wall temperature t_w can now be calculated by Eq. (45):

$$t_w = \frac{2600 \times 201 + 750 \times 239}{2600 + 750}$$
$$= 209°F.$$

Since this value does not agree with the value initially assumed, the calculations must be repeated.

As a second trial, assume $t_w = 220$°F. From Fig. 38,

$$h = 800 \text{ B.t.u./(sq. ft.)(hr.)(°F.)} \qquad (\Delta t = 220 - 201 = 19°F.);$$

so

$$h_1 = 1.25 \times 800 \times 0.94$$
$$= 940 \text{ B.t.u./(sq. ft.)(hr.)(°F.).}$$

By Eq. (24),

$$4,400,000 = 940 \times A_1 \times (220 - 201)$$

or

$$A_1 = 246 \text{ sq. ft.}$$

The corresponding outside tube surface

$$A_2 = 246 \times \left(\frac{2.000}{1.834}\right) = 269 \text{ sq. ft.}$$

Hence, $W = 4,620/269 = 17.2$ lb./(sq. ft.)(hr.). Since

$$W \times N = 17.2 \times 5 = 86,$$

method b of Case 22 must be used to determine h_2. From Fig. 36,

$$h_0 = 2000 \text{ B.t.u./(sq. ft.)(hr.)(°F.);}$$

from Table 37, $F_t = 1\ 56$ (at $t_a = 220°F$), and from Table 43, $F_{w}'' = 1\ 27$ Hence, for clean tubes,

$$h_t = 0\ 29 \times 2{,}000 \times 1\ 56 \times 1\ 27 = 1150 \text{ B t u /(sq ft)(hr)(°F.)}$$

Combining this with a scale coefficient $h_s = 3000$ B t u /(sq ft)(hr)(°F),

$$\frac{1}{h_t'} = \frac{1}{1150} + \frac{1}{3000},$$

or

$$h_t' = 830 \text{ B t u /(sq ft)(hr)(°F)}$$

Calculating the tube-wall temperature t_w by Eq (45),

$$t_w = \frac{910 \times 201 + 830 \times 239}{910 + 830}$$
$$= 219°F$$

Since this is approximately the value initially assumed, the calculations need not again be repeated

The outside tube surface $A_2 = 260$ sq ft , determined above, does not take into account the temperature drop through the tube wall A more accurate value can be calculated as follows Since the tube walls are 0 083 in thick and the thermal conductivity of brass at 220°F is about

$$59 \text{ B t u /(ft)(hr)(°F)},$$

by Eq (27),

$$\frac{1}{U_2} = \frac{2\ 000}{910 \times 1\ 834} + \frac{0\ 083}{12 \times 59} + \frac{1}{830}$$

or

$$U_2' = 400 \text{ B t u /(sq ft)(hr)(°F)}$$

Hence, by Eq (24),

$$4{,}400{,}000 = 400 \times A_2 \times (239 - 201),$$

or

$$A_2 = 290 \text{ sq ft}$$

Since 1 910 lin ft of 2-in tubing has 1 sq ft of outside surface, the outside surface of each tube = 5/1 910 = 2 62 sq ft , and the total number of tubes required = 290/2 62 = 111 tubes

Illustrative Problem 7—Calculate the heat loss per foot of length from a bare 3-in steam pipe at 400°F if the surrounding air and walls are at 70°F

Solution—The convection coefficient h can be determined by Case 12 Thus, from Fig 32,

$$h_c = 2\ 1 \text{ B t u /(sq ft)(hr)(°F)} \qquad \text{(for } \Delta t = 400 - 70 = 330°F \text{),}$$

from Table 28, $F_t = 0\ 98$ (film temperature = $(400 + 70)$ 2 = 235°F), from Table 29, $F_d = 0\ 73$ (outside pipe diameter = 3 500 in), and from Table 30, $F_p = 1\ 00$ Hence,

$$h = 2\ 1 \times 0\ 98 \times 0\ 73 \times 1\ 00$$
$$= 1\ 5 \text{ B t u /(sq ft)(hr)(°F)}$$

The radiation coefficient h_r can be determined from Fig. 10 and is equal to 2.4 B.t.u./(sq. ft.)(hr.)(°F.). Assuming that the emissivity of the pipe is 0.9, the combined coefficient for convection and radiation is

$$1.5 + 0.9 \times 2.4 = 3.7 \text{ B.t.u.}/(\text{sq. ft.})(\text{hr.})(°F.).$$

Therefore, since 1.091 ft. of 3-in. pipe have 1 sq. ft. of outside surface, the total heat loss from the pipe

$$= 3.7 \times \left(\frac{1}{1.091}\right) \times (400 - 70) = 1120 \text{ B.t.u.}/(\text{hr.})(\text{ft. of length}).$$

Case 1. Liquids Heated inside Horizontal or Vertical Tubes, Turbulent Flow.—The film coefficient can be determined by the equation

$$h = h_a \times F_t \times F_d$$

where h = the film coefficient, Btu /(sq ft)(hr)(°F); h_a = the base value of the film coefficient, from Fig 22, F_t = the temperature-correction factor, from Table 8, F_d = the diameter-correction factor, from Table 9

TABLE 8—TEMPERATURE-CORRECTION FACTOR FOR CASE 1

Liquid	Ave temp of liquid					
	0°F	50°F	100°F	150°F	200°F	250°F
			F_t			
Acetic acid (100%)			1 00	1 04	1 08	1 12
Acetic acid (50%)		0 75	1 00	1 15	1 30	1 46
Acetone	0 78	0 91	1 00	1 02	1 04	1 06
Ammonia	0 69	0 84	1 00	1 13	1 25	1 35
Amyl acetate	0 96	0 98	1 00	1 05	1 15	1 27
Amyl alcohol (iso)	0 41	0 68	1 00	1 39	1 82	2 25
Aniline		0 75	1 00	1 31	1 70	2 10
Benzene		0 80	1 00	1 14	1 28	1 42
Brine (CaCl₂) (25%)	0 54	0 74	1 00	1 29	1 52	2 01
Butyl alcohol (n)	0 50	0 72	1 00	1 33	1 72	2 13
Carbon disulfide	0 91	0 95	1 00	1 03	1 05	1 06
Carbon tetrachloride	0 73	0 88	1 00	1 05	1 10	1 18
Chlorobenzene	0 88	0 93	1 00	1 02	1 04	1 05
Chloroform	0 75	0 86	1 00	1 11	1 41	1 47
Ethyl acetate	1 01	1 01	1 00	0 99	0 98	0 97
Ethyl alcohol (100%)	0 63	0 80	1 00	1 21	1 43	1 84
Ethyl alcohol (40%)	0 59	0 76	1 00	1 41	1 80	2 40
Ethyl bromide	0 87	0 95	1 00	1 04	1 08	1 11
Ethylene glycol (50%)	0 45	0 69	1 00	1 40	1 69	2 40
Ethyl ether	0 81	0 91	1 00	1 05	1 12	1 18
Ethyl iodide	0 70	0 85	1 00	1 14	1 20	1 44
Glycerol (50%)	0 45	0 69	1 00	1 39	1 80	2 80
Heptane	0 81	0 92	1 00	1 08	1 18	1 40
Hexane	0 84	0 92	1 00	1 07	1 18	1 38
Methyl alcohol (100%)	0 66	0 87	1 00	1 10	1 18	1 28
Methyl alcohol (90%)	0 53	0 75	1 00	1 30	1 61	2 38
Methyl alcohol (40%)	0 33	0 57	1 00	1 30	1 61	1 91
Octane (n)	0 84	0 92	1 00	1 07	1 15	1 10
Pentane (n)	0 85	0 93	1 00	1 04	1 07	1 10
Propyl alcohol (iso)	0 36	0 69	1 00	1 32	1 64	1 93
Sulfur dioxide	0 96	0 98	1 00	1 02	1 07	1 11
Sulfuric acid (60%)			0 91	1 00	1 19	1 52
Toluene	0 80	0 90	1 00	1 10	1 21	1 31
Water			0 90	1 00	1 23	1 41

* Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 9—DIAMETER-CORRECTION FACTOR FOR CASE 1

Inside tube dia in	F_d	Inside tube dia in	F_d
0 20	1 38	1 20	0 97
0 30	1 27	1 30	0 91
0 40	1 20	1 40	0 94
0 50	1 15	1 50	0 82
0 60	1 11		
0 70	1 08	2 00	0 77
0 80	1 05	2 50	0 83
0 90	1 02	3 00	0 80
1 00	1 00	3 50	0 78
1 10	0 98	4 00	0 76

TABLE 10—MINIMUM PERMISSIBLE VALUES OF $V \times d$ FOR CASE 1

V = velocity, ft per sec × d = inside tube dia in

Liquid	Ave temp of liquid				
	50°F	100°F	150°F	200°F	250°F
	$(V \times d)$ min.				
Acetic acid (100%)		1 2	0 9	0 8	0 7
Acetic acid (50%)	3 5	1 7	1 0	0 6	0 5
Amyl acetate	1 6	1 0	0 8	0 4	0 3
Amyl alcohol (iso)	9 3	4 1	2 0	1 2	0 8
Aniline	7 6	3 7	1 9	1 0	0 7
Brine (CaCl₂) (25%)	1 9	1 1	0 7	0 4	0 2
Butyl alcohol (n)	6 2	2 9	1 5	1 0	0 6
Ethyl alcohol (100%)	2 8	1 5	0 8	0 5	0 3
Ethyl alcohol (40%)	5 8	2 3	1 2	0 8	0 7
Ethylene glycol (50%)	8 3	2 8	1 2	0 7	0 4
Glycerol (50%)	9 7	4 2	2 3	1 3	0 8
Methyl alcohol (40%)	3 4	1 5	0 8	0 5	0 4
Propyl alcohol (iso)	3 4	2 3	1 2	0 8	0 8
Sulfuric acid (60%)	6 8	3 4	2 2	1 8	1 4

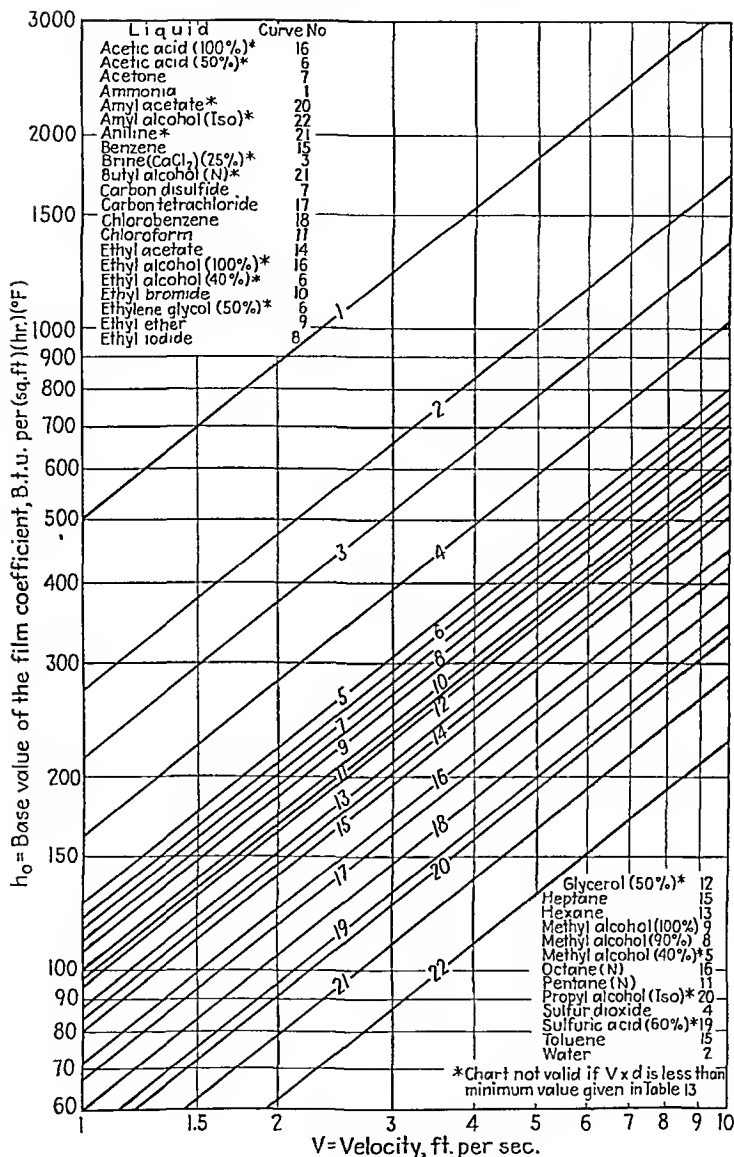* Only those liquids for which $(V \times d)$ min. at 100°F is greater than 0 2 are listed.
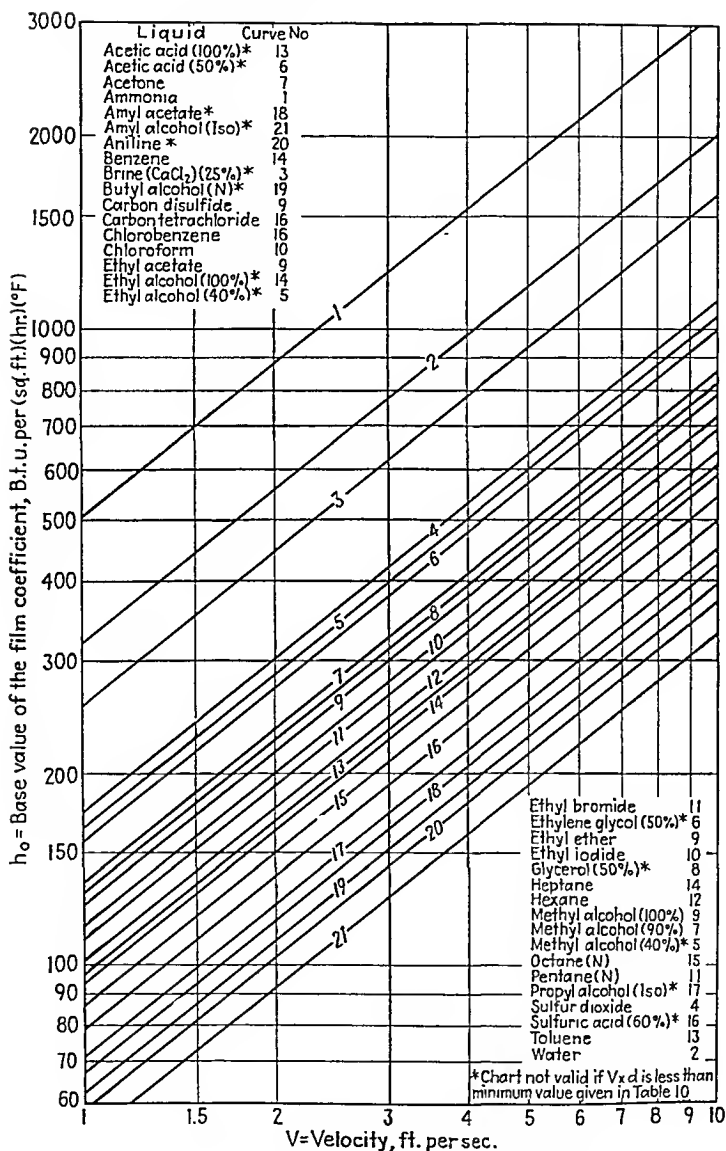
FIG. 22.—Base value of the film coefficient for Case 1.

Case 2. Liquids Cooled inside Horizontal or Vertical Tubes, Turbulent Flow.—The film coefficient can be determined by the equation

$$h = h_a \times F_t \times F_d$$

where h = the film coefficient, B t u f/(sq ft)(hr)(°F), h_a = the base value of the film coefficient, from Fig. 23, F_t = the temperature-correction factor, from Table 11; F_d = the diameter-correction factor, from Table 12

TABLE 11—TEMPERATURE CORRECTION FACTOR FOR CASE 2

Liquid	Ave temp of liquid					
	0°F	50°F	100°F	150°F	200°F	250°F
				F_t		
Acetic acid (100%)			1 00	1 04	1 09	1 13
Acetic acid (50%)		0 70	1 00	1 26	1 47	1 65
Acetone	0 75	0 89	1 00	1 03	1 05	1 07
Ammonia	0 62	0 75	1 00	1 88	1 87	1 81
Amyl acetate	0 92		1 00	1 02		1 05
Amyl alcohol (iso)		0 63	1 00	1 47	2 01	2 55
Aniline		0 70	1 00	1 40	1 92	2 43
Benzene		0 78	1 00	1 18	1 33	1 49
Brine (CaCl₂) (25%)	0 48	0 70	1 00	1 33	1 63	2 30
Butyl alcohol (n)	0 47	0 69	1 00	1 40	1 88	2 35
Carbon disulfide	0 83	0 92	1 00	1 04	1 08	1 10
Carbon tetrachloride	0 69	0 83	1 00	1 07	1 15	1 17
Chlorobenzene	0 90	0 95	1 00	1 03	1 07	1 08
Chloroform	0 79	0 89	1 00	1 11	1 21	1 32
Ethyl acetate	0 99	1 00	1 00	1 00	0 99	0 98
Ethyl alcohol (100%)	0 58	0 77	1 00	1 25	1 52	1 80
Ethyl alcohol (40%)		0 58	1 00	1 45	1 90	2 3a
Ethyl bromide	0 86	0 93	1 00	1 05	1 10	1 14
Ethylene glycol (50%)	0 37	0 60	1 00	1 53	2 13	
Ethyl ether	0 79	0 91	1 00	1 08	1 16	1 a4
Ethyl iodide	0 69	0 8a	1 00	1 1a	1 a0	1 a3
Glycerol (50%)	0 37	0 63	1 00	1 4a	1 90	2 3a
Heptane	0 87	0 91	1 00	1 09	1 18	1 a8
Hexane	0 81	0 90	1 00	1 09	1 17	1 a5
Methyl alcohol (100%)	0 68	0 84	1 00	1 17	1 34	1 50
Methyl alcohol (90%)	0 61	0 82	1 00	1 14	1 a	1 39
Methyl alcohol (40%)	0 50	0 63	1 00	1 33	1 66	1 90
Octane (n)	0 79	0 90	1 00	1 09	1 16	1 a4
Pentane (n)	0 86	0 93	1 00	1 05	1 10	1 a4
Propyl alcohol (iso)		0 63	1 00	1 39	1 80	2 2a
Sulfur dioxide	0 93	0 96	1 00	1 03	1 07	1 10
Sulfuric acid (60%)			1 00	1 a3	2 ae	1 43
Toluene	0 76	0 85	1 00	1 11	1 20	1 a7
Water		0 6a	1 00	1 30	1 56	1 7*

*Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 12—DIAMETER-CORRECTION FACTOR FOR CASE 2

Inside tube dia in	F_d	Inside tube dia in	F_d
0 20	1 38	1 20	0 97
0 30	1 27	1 30	0 9a
0 40	1 20	1 40	0 9a
0 50	1 15	1 50	0 92
0 60	1 11		
0 70	1 08	2 00	0 87
0 80	1 0a	2 50	0 83
0 90	1 02	3 00	0 80
1 00	1 00	3 50	0 78
1 10	0 98	4 00	0 76

TABLE 13—MINIMUM PERMISSIBLE VALUES OF $V \times d$ FOR CASE 2

V = velocity, ft per sec, d = inside tube dia, in

Liquid*	Ave temp of liquid				
	50°F	100°F	150°F	200°F	250°F
			($V \times d$) min		
Acetic acid (100%)		1 2	0 9	0 8	0 7
Acetic acid (50%)	3 5	1 7	1 0	0 7	0 6
Amyl acetate	1 8	1 0	0 6	0 4	0 3
Amyl alcohol (iso)	9 5	4 1	2 0	1 1	0 6
Aniline	7 6	3 7	1 9	1 0	0 7
Benzene	1 9	1 3	0 9	0 7	0 5
Butyl alcohol (n)	6 2	3 2	1 7	0 9	0 6
Ethyl alcohol (100%)	2 1	1 5	1 0	0 8	0 5
Ethyl alcohol (40%)	5 6	2 3	1 2	0 8	0 7
Ethylene glycol (50%)	6 3	2 6	1 3	0 8	0 7
Glycerol (50%)	9 7	4 2	2 3	1 2	0 8
Methyl alcohol (40%)	3 3	1 4	0 8	0 8	0 4
Propyl alcohol (iso)	5 4	2 3	1 2	0 8	0 6
Sulfuric acid (60%)	6 5	3 4	2 2	1 8	1 7

* Only those liquids for which ($V \times d$) min at 100°F is greater than 0 2 are listed.

Liquid	Curve No
Acetic acid (100%)*	16
Acetic acid (50%)*	6
Acetone	7
Ammonia	1
Amyl acetate*	20
Amyl alcohol (Iso)*	22
Aniline*	21
Benzene	15
Brine (CaCl₂) (25%)*	3
Butyl alcohol (N)*	21
Carbon disulfide	7
Carbon tetrachloride	17
Chlorobenzene	18
Chloroform	11
Ethyl acetate	14
Ethyl alcohol (100%)*	16
Ethyl alcohol (40%)*	6
Ethyl bromide	10
Ethylene glycol (50%)*	6
Ethyl ether	9
Ethyl iodide	8

Glycerol (50%)*	12
Heptane	15
Hexane	13
Methyl alcohol (100%)	9
Methyl alcohol (90%)	8
Methyl alcohol (40%)*	5
Octane (N)	16
Pentane (N)	11
Propyl alcohol (Iso)*	20
Sulfur dioxide	4
Sulfuric acid (60%)*	19
Toluene	15
Water	2

*Chart not valid if V x d is less than minimum value given in Table 13

h_o = Base value of the film coefficient, B.t.u. per (sq.ft.)(hr.)(°F)

V = Velocity, ft. per sec.

Fig. 23.—Base value of the film coefficient for Case 2.

Case 3. Gases Heated or Cooled inside Horizontal or Vertical Tubes, Turbulent Flow.—The film coefficient can be determined by the equation

$$h = h_a \times F_t \times F_d$$

where h = the film coefficient, B t u /(sq ft)(hr)(°F)
h_a = base value of the film coefficient, from Fig 24
F_t = the temperature-correction factor, from Table 14
F_d = the diameter-correction factor, from Table 15

TABLE 14—TEMPERATURE-CORRECTION FACTOR FOR CASE 3

Gas	Ave temp of gas						
	−100°F	0°F	100°F	200°F	300°F	400°F	500°F
	F_t						
Acetone			1 00	1 15	1 32		1 59
Acetylene		0 92	1 00	1 08	1 15		1 22
Air	0 99	0 06	1 00	1 04	1 07	1 09	1 12
Ammonia	0 75	0 01	1 00	1 07	1 13	1 18	1 22
Benzene			1 00	1 29	1 44		1 64
Butane		0 81	1 00	1 09	1 17		1 22
Carbon dioxide	0 82	0 92	1 00	1 05	1 11	1 16	1 22
Carbon monoxide	0 81	0 9a	1 00	1 0a	1 09	1 15	1 17
Chlorine	0 81	0 00	1 00	1 03	1 05	1 08	1 11
Chloroform			1 00	1 09	1 19	1 29	1 38
Ethane	0 73	0 87	1 00	1 18	1 35	1 51	1 66
Ethyl acetate			1 00	1 18	1 35	1 51	1 66
Ethyl alcohol			1 00	1 04	1 08	1 12	1 16
Ethyl chloride		0 88	1 00	1 07	1 11	1 15	1 18
Ethylene	0 75	0 88	1 00	1 11	1 22	1 33	1 44
Ethyl ether		0 89	1 00	1 12	1 25	1 41	1 50
Helium	0 94	0 97	1 00	1 03	1 06	1 08	1 13
Hydrogen	0 91	0 96	1 00	1 04	1 08	1 12	1 13
Hydrogen sulfide	0 85	0 92	1 00	1 07	1 14	1 20	1 26
Methane	0 83	0 93	1 00	1 0a	1 09	1 13	1 16
Methyl chloride	0 69	0 83	1 00	1 16	1 31	1 46	1 60
Nitric oxide	0 91	0 96	1 00	1 03	1 06	1 08	1 10
Nitrogen	0 93	0 98	1 00	1 03	1 05	1 08	1 10
Nitrous oxide	0 96	0 98	1 00	1 02	1 04	1 06	1 08
Oxygen	0 91	0 98	1 00	1 03	1 06	1 08	1 10
Pentane (iso)			1 00	1 14	1 31		
Steam			1 00	1 03	1 05	1 10	1 14
Sulfur dioxide		0 94	1 00	1 0a	1 10	1 14	1 18

* Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 15—DIAMETER CORRECTION FACTOR FOR CASE 3

Inside Din	F_d
0 20	1 38
0 30	1 27
0 40	1 20
0 50	1 15
0 60	1 11
0 70	1 08
0 80	1 05
0 90	1 02
1 00	1 00
1 10	0 98
1 20	0 97
1 30	0 95
1 40	0 94
1 50	0 92
2 00	0 87
2 50	0 83
3 00	0 80
3 50	0 78
4 00	0 76

TABLE 16—MINIMUM PERMISSIBLE VALUES OF $G = V_p$ FOR CASE 3

$$G = \frac{\text{rate of flow}}{\text{cross-sectional area}} \quad \frac{\text{lb per sec}}{\text{sq ft}}$$

V = velocity, ft per sec
ρ = density, lb per cu ft

Pressure Lb per Sq In Gauge	$G_{min} = (V\rho)_{min}$
0	0 46
50	1 00
100	1 73
150	2 18
200	2 59
250	2 97
300	3 32
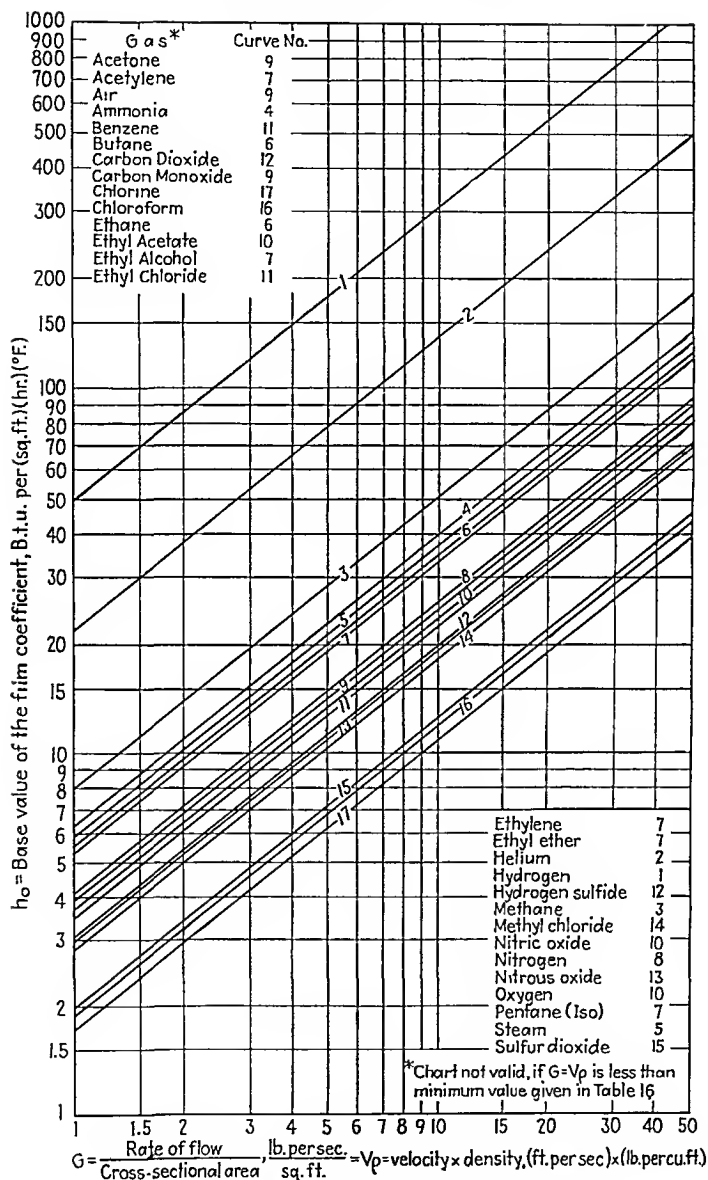
Fig. 24.—Base value of the film coefficient for Case 3.

Case 4. Gases at Atmospheric Pressure Heated or Cooled inside Horizontal or Vertical Tubes, **Turbulent Flow** —The film coefficient can be determined by the equation

$$h = h_b \times F_t \times F_d$$

where h = the film coefficient, B t u /(sq ft)(hr)(°F)
h_b = the base value of the film coefficient, from Fig 25
F_t = the temperature-correction factor, from Table 17
F_d = the diameter-correction factor, from Table 18.

TABLE 17 —TEMPERATURE-CORRECTION FACTOR FOR CASE 4

Gas	Ave temp of gas						
	−100°F	0°F	100°F	200°F	300°F	400°F	500°F
	F_t*						
Acetylene	*1.80*	1.12	1.00	0.91	0.84	0.76	0.69
Air	1.29	1.14	1.00	0.91	0.84	0.76	0.73
Ammonia		1.03	1.00	0.94	0.87	0.80	0.72
Butane			1.00	0.96	0.93	0.91	0.89
Carbon dioxide	1.17	1.08	1.00	0.93	0.87	0.83	0.79
Carbon monoxide	*1.2*	1.12	1.00	0.89	0.81	0.75	0.68
Chlorine		1.1*	1.00	0.91	0.8*	0.74	0.68
Ethane	1.04	1.0*	1.00	0.98	0.9	0.86	0.8a
Ethylene	1.07	1.03	1.00	0.94	0.8*	0.86	0.8a
Helium	1.53	1.14	1.00	0.89	0.9	0.7*	0.63
Hydrogen	1.30	1.13	1.00	0.90	0.81	0.78	0.64
Hydrogen sulfide		1.09	1.00	0.94	0.59	0.8*	0.81
Methane	1.19	1.09	1.00	0.9*	0.84	0	0
Methyl chloride		0.9*	1.00	1.02	1.03	1.04	1.05
Nine oxide	1.30	1.1*	1.00	0.9*	0.87	0*	0.6*
Nitrogen	1.30	1.13	1.00	0.90	0.81	0*	0.64
Nitrous oxide	1.36	1.14	1.00	0.83	0.8	0*	0.60
Oxygen	1.30	1.12	1.00	0.90	0.84	0.4	0.6*
Sulfur dioxide			1.00	0.92	0.86	0.81	0.7*

*Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 18 —DIAMETER CORRECTION FACTOR FOR CASE 4

Inside Tube Dia. in	F_d
0.20	1.34
0.30	1.27
0.40	1.20
0.50	1.15
0.60	1.11
0.70	1.08
0.80	1.05
0.90	1.02
1.00	1.00
1.10	0.98
1.20	0.97
1.30	0.95
1.40	0.94
1.50	0.92
2.00	0.87
2.50	0.83
3.00	0.80
3.50	0.78
4.00	0.76

TABLE 19 —MINIMUM PERMISSIBLE VALUES OF V FOR CASE 4

V = velocity ft per sec

Gas*	Ave temp of gas					
	0°F	100°F	200°F	300°F	400°F	500°F
	ft per sec					
Ammonia	9	11	13	15	17	19
Helium	39	47	55	64	72	81
Hydrogen	75	9*	108	1*3	141	15*
Methane	10	12	14	16	18	20

* Only those gases for which V_{min} is greater than 10 ft per sec at temperatures up to 500 F are listed.
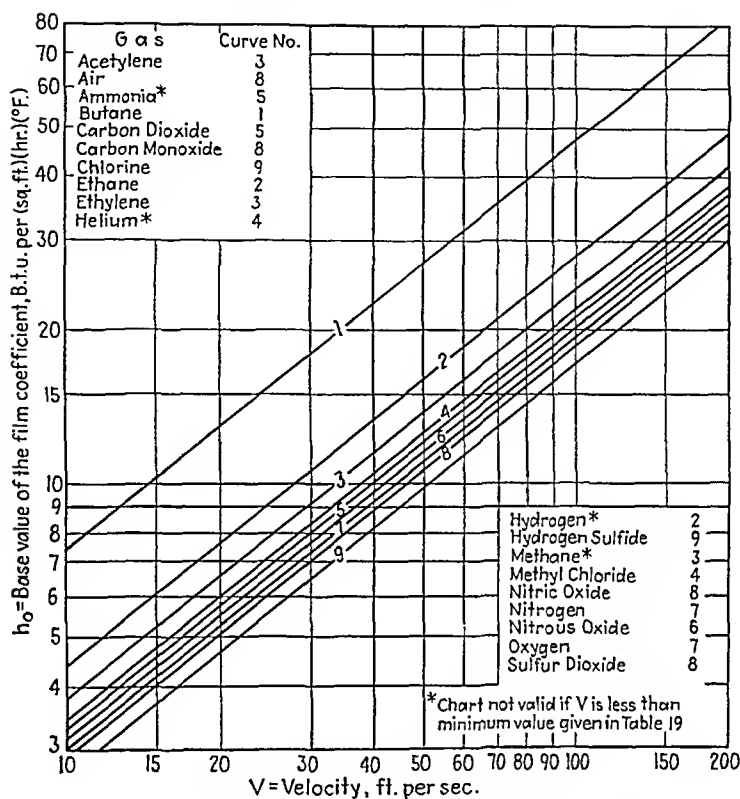
FIG. 25.—Base value of the film coefficient for Case 4.

Case 5 **Liquids Heated or Cooled outside Single Tubes, Direction of Flow Normal to Tube**—The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d$$

where h = the film coefficient, B t u /(sq ft)(hr)(°F)

h_b = the base value of the film coefficient from Fig 26

F_t = the temperature-correction factor, from Table 20

F_d = the diameter-correction factor, from Table 21

TABLE 20—TEMPERATURE-CORRECTION FACTOR FOR CASE 5

Liquid	Ave temp of the film*					
	0°F	50°F	100°F	150°F	200°F	250°F
	F_t					
Acetic acid (100%)					0 98	0 89
Acetic acid (50%)		0 82	1 00	1 12	1 19	1 23
Acetone	0 50	0 95	1 00	1 01	1 03	1 05
Ammonia	0 79	0 89	1 00	1 11	1 23	1 34
Amyl acetate	1 16	1 08	1 00	0 93	0 85	0 78
Amyl alcohol (iso)	0 57	0 77	1 00	1 24	1 48	1 72
Aniline		0 84	1 00	1 20	1 42	1 63
Benzene	0 93	0 97	1 00	1 08	1 16	1 24
Brine (CaCl₂) (25%)	0 63	0 80	1 00	1 21	1 47	1 70
Butyl alcohol (n)	0 53	0 89	1 00	1 21	1 43	1 55
Carbon disulfide	0 97	0 98	1 00	1 01	1 02	1 03
Carbon tetrachloride		0 92	1 00	1 01	1 02	1 03
Chlorobenzene	1 06	1 03	1 00	0 97	0 95	0 94
Chloroform	0 88	0 94	1 00	1 02	1 09	1 15
Ethyl acetate	1 18		1 00	0 94	0 87	0 81
Ethyl alcohol (100%)	0 74	0 87	1 00	1 13	1 25	1 36
Ethyl alcohol (40%)	0 47	0 72	1 00	1 28	1 55	1 77
Ethyl bromide	0 93	0 97	1 00	1 02	1 03	1 04
Ethylene glycol (50%)	0 58	0 74	1 00	1 28	1 55	1 84
Ethyl ether	0 91	0 95	1 00	1 03	1 09	1 15
Ethyl iod de	0 78	0 91	1 00	1 10	1 19	1 27
(lycerol (50%))	0 29	0 77	1 00	1 53	1 73	
Heptane	0 91	0 94	1 00	1 03	1 06	1 09
Hexane	0 81	0 95	1 00	1 01	1 07	1 10
Methyl alcohol (100%)	0 79	0 91	1 00	1 06	1 10	1 13
Methyl alcohol (40%)	0 78	0 89	1 00	1 08	1 14	1 20
Methyl alcohol (40%)	0 60	0 78	1 00	1 20	1 36	1 40
Octane (n)	0 81	0 96	1 00	0a	1 03	1 13
Pentane (n)	0 94	0 97	1 00	1 02	1 03	1 04
I ropyl alcohol (iso)	0 53	0 77	1 00	1 21	1 38	1 50
Sulfur dioxide	1 03	1 01	1 00	0 99	0 99	0 98
Sulfuric acid (60%)	0 90	0 80	1 00	1 09	1 19	1 88
Toluene	0 90	0 95	1 00	1 04	1 07	1 09
Water		0 77	1 00	1 16	1 30	1 41

* Average temperature of the film = (temperature of the tube wall + temperature of the main body of the liquid) ÷ 2

† Values in italics are from extrapolated values of the physical properties of the fluids

TABLE 21—DIAMETER CORRECTION FACTOR FOR CASE 5

Outs de of Tube D in	F_d
0 250	1 64
0 375	1 54
0 500	1 36
0 625	1 23
0 750	1 14
1 00	1 00
1 25	0 91
1 50	0 84
2 00	0 74
2 50	0 67
3 00	0 62
3 50	0 58
4 00	0 54

Liquid	Curve No
Acetic acid (100%)	13
Acetic acid (50%)	5
Acetone	9
Ammonia	1
Amyl acetate	19
Amyl alcohol (Iso)	20
Aniline	16
Benzene	14
Brine (CaCl$_2$)(25%)	3
Butyl alcohol (N)	17
Carbon disulfide	10
Carbon tetrachloride	17
Chlorobenzene	18
Chloroform	12
Ethyl acetate	14
Ethyl alcohol(100%)	12
Ethyl alcohol(40%)	4

Ethyl bromide	13
Ethylene glycol (50%)	4
Ethyl ether	11
Ethyl iodide	13
Glycerol (50%)	6
Heptane	14
Hexane	13
Methyl alcohol (100%)	9
Methyl alcohol(90%)	8
Methyl alcohol(40%)	5
Octane(N)	15
Pentane(N)	12
Propyl alcohol (Iso)	16
Sulfur dioxide	7
Sulfuric acid(60%)	14
Toluene	13
Water	2

FIG. 26.—Base value of the film coefficient for Case 5.

Case 7. Gases at Atmospheric Pressure Heated or Cooled outside Single Tubes, Direction of Flow Normal to Tube —The film coefficient can be determined by the equation

$$h = h_o \times F_t \times F_d$$

where h = the film coefficient, B t u /(sq ft)(hr)(°F)

h_o = the base value of the film coefficient, from Fig 28

F_t = the temperature-correction factor, from Table 24

F_d = the diameter correction factor from Table 25

Note that the film coefficient h obtained from the foregoing equation is for heat transfer by *convection* only   If the heat-transfer surface is exposed to other surfaces at different temperatures, it will gain or lose heat by *radiation* also   The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface, obtained from Table IX in the Appendix, times the radiation coefficient h_r, obtained from Fig 10 in Chap II, onto the convection coefficient h obtained from the equation above

TABLE 24 —TEMPERATURE CORRECTION FACTOR FOR CASE 7

Gas	Ave temp of the film*						
	−100°F †	0°F	100°F	200°F	300°F	400°F	500°F
				F †			
Acetylene		0 97	1 00	1 03	1 05	1 07	1 07
Air	1 07	1 03	1 00	0 97	0 94	0 91	0 90
Ammonia		0 94	1 00	1 01	1 03	1 05	1 04
Butane			1 00	1 0₀	1 10	1 13	1 16
Carbon dioxide	0 94	0 98	1 00	1 01	1 01	1 02	1 02
Carbon monoxide	1 0₄	1 02	1 00	0 98	0 97	0 95	0 93
Chlorine		1 01	1 00	0 98	0 96	0 94	0 93
Ethane	0 86	0 91	1 00	1 08	1 16	1 24	1 33
Ethylene	0 82	0 93	1 00	1 00	1 12	1 18	1 25
Helium	1 13	1 06	1 00	0 95	0 90	0 86	0 82
Hydrogen	1 03	1 04	1 00	0 96	0 93	0 90	0 86
Hydrogen sulfide		1 01	1 00	0 98	0 97	0 95	0 93
Methane	1 09	1 04	1 00	0 97	0 91	0 92	0 90
Methyl chloride		0 87	1 00	1 11	1 20	1 29	1 38
Nitric oxide	1 00	1 03	1 00	0 97	0 95	0 93	0 90
Nitrogen	1 09	1 04	1 00	0 98	0 95	0 90	0 86
Nitrous oxide	1 10	1 08	1 00	0 94	0 88	0 83	0 79
Oxygen	1 07	1 03	1 00	0 97	0 94	0 9¹	0 90
Sulfur dioxide			1 00	1 00	0 99	0 98	0 98

* Average temperature of the film = (tempeıat ıre of the tube wall + temperature of the ı ain bo ly of the gas) ÷ 2
† Values in ıtal cs aıe based on extrapolated values of the physıcal properties of the flu ıs

TABLE 25 —DIAMETER-CORRECTION FACTOR FOR CASE 7

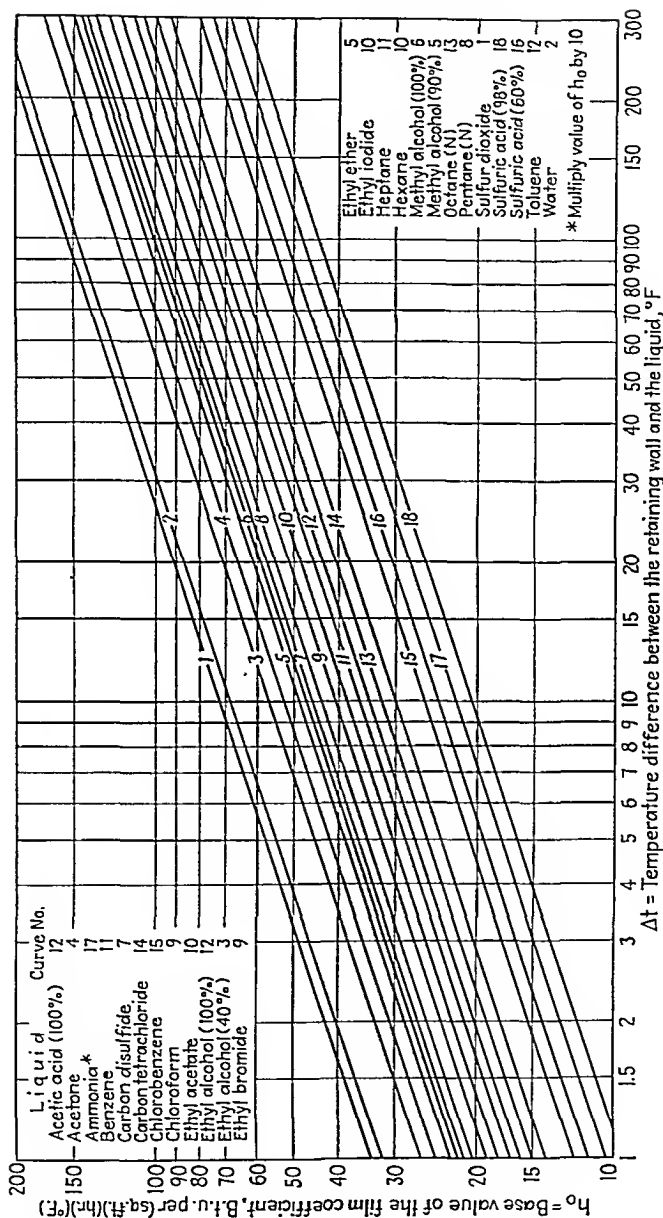| Outs ıe In	
D ₐ In	F_d
0 250	1 84
0 375	1 51
0 500	1 30
0 625	1 23
0 7₅₀	1 1₄
1 00	1 00
1 25	0 9₁
1 50	0 8₄
2 00	0 74
2 50	0 67
3 00	0 62
3 50	0 58
4 00	0 5₄
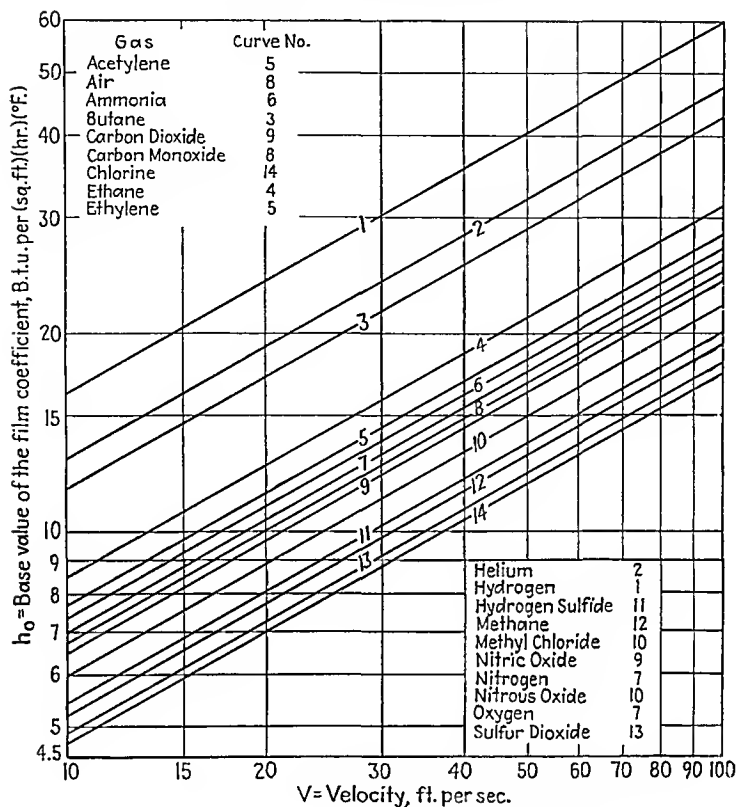
FIG. 28.—Base value of the film coefficient for Case 7.

Case 8   Liquids or Gases Heated or Cooled outside Tube Bundles, Direction of Flow Normal to Tubes—Approximate values of the film coefficients for fluids flowing outside tube bundles and in a direction normal to the tubes can be determined by the following equations $h = 1 3 \times h_{sl}$ for *staggered* tubes or $h = 1 2 \times h_{sl}$ for tubes in *line* where $h =$ the film coefficient for flow normal to the tube bundle, B t u /(sq ft)(hr)(°F), and $h_{sl} =$ the film coefficient for flow normal to a single tube, obtained from Case 5 6, or 7 h_{sl} should be evaluated at a velocity equal to that at the narrowest section between the tubes of the tube bundle

For baffled heat exchangers in which the flow is normal to most of the tubes but parallel to some it is suggested that the film coefficient for the shell side fluid be taken equal to h_{sl}, the latter being evaluated at the velocity

$$V = \frac{144Q}{(d_s - nd)l}$$

where $V =$ the velocity between the tubes at the widest part of the shell, ft per sec , $Q =$ the volumetric rate of flow cu ft per sec , $d_s =$ the inside diameter of the shell in , $d =$ the outside diameter of the tubes in , $n =$ the number of tubes across the widest part of the shell, $l =$ the distance between baffles in

Case 9   Liquids or Gases Heated or Cooled outside Tube Bundles, Direction of Flow Parallel to Tubes—The film coefficient for fluids flowing *outside* tube bundles and in a direction parallel to the tubes can be determined from Case 1 2 3 or 4 for fluids *inside* tubes if an equivalent inside diameter d_e is used in determining the diameter correction factor F_d The equivalent diameter can be calculated by the equation

$$d_e = 4 \times \frac{A}{P} \tag{46}$$

where $d_e =$ the equivalent inside diameter in , $A =$ the cross sectional area of the space between the tubes sq in , $P =$ the number of tubes $\times$ the perimeter of each tube in

For tubes spaced on equilateral triangles or on squares d_e can be determined from Fig 29 or 30

Case 10   Liquids or Gases Heated or Cooled in Annular Spaces, Turbulent Flow—The film coefficients for fluids flowing through the *annular space* of a double-pipe heat exchanger can be determined from Case 1 2 3, or 4 for fluids *inside* tubes if an equivalent inside diameter d_e is used in determining the diameter correction factor F_d The equivalent diameter can be calculated by the equation

$$d_e = \frac{d_1{}^2 - d_2{}^2}{d_2} \tag{47}$$

where $d_e =$ the equivalent inside diameter in , $d_1 =$ the inside diameter of the outer pipe in , $d_2 =$ the outside diameter of the inner pipe in
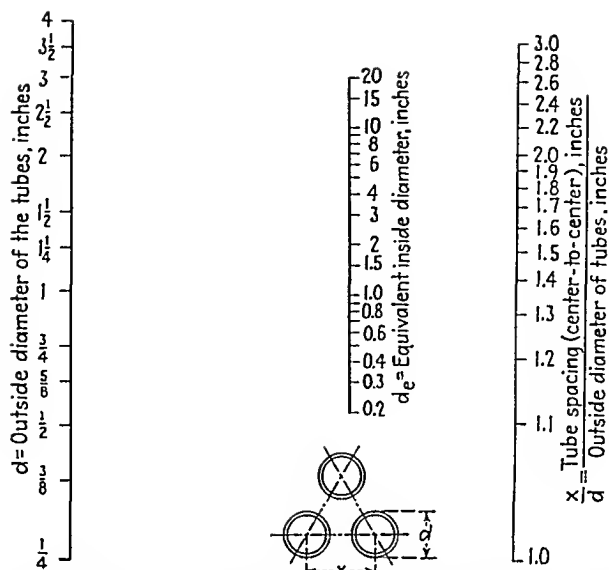
Fig. 29—Alignment chart for determining the equivalent inside diameter for parallel-flow outside tube bundles, tubes spaced on equilateral triangles.
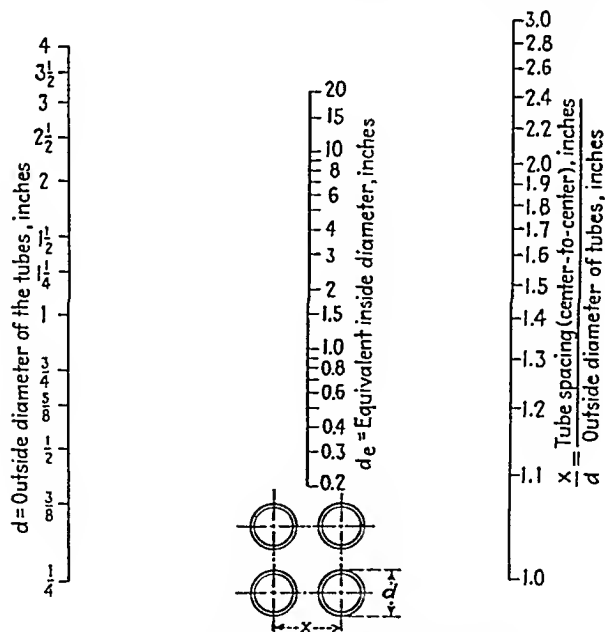
Fig. 30.—Alignment chart for determining the equivalent inside diameter for parallel flow outside tube bundles, tubes spaced on squares.

Case 11. Liquids Heated outside Single Horizontal Tubes, Natural Convection.—The film coefficient can be determined by the equation

$$h = h_b \times F_t \times F_d$$

where h = the film coefficient, B t u /(sq ft)(hr)(°F)

h_b = the base value of the film coefficient, from Fig 31

F_t = the temperature-correction factor, from Table 26

F_d = the diameter correction factor, from Table 27

TABLE 26.—TEMPERATURE CORRECTION FACTOR FOR CASE 11

Liquid	Ave temp of the film[a]					
	0°F	50°F	100°F	150°F	200°F	250°F
	F_t[b]					
Acetic acid (100%)			1 00	0 95	0 90	0 87
Acetone		0 97	1 00	1 01	1 02	1 03
Ammonia	0 79	0 89	1 00	1 11	1 22	1 33
Benzene		0 91	1 00	1 08	1 16	1 23
Carbon disulfide	0 97	0 98	1 00	1 01	1 02	1 03
Carbon tetrachloride		0 95	1 00	0 99	0 98	0 96
Chlorobenzene	1 11	1 04	1 00	0 95	0 91	0 88
Chloroform	0 88	0 95	1 00	1 09		1 14
Ethyl acetate	1 17	1 08	1 00	0 92	0 85	0 78
Ethyl alcohol (100%)	0 76	0 88	1 00	1 12	1 23	1 33
Ethyl alcohol (40%)	0 49	0 73	1 00	1 27		
Ethyl bromide	0 95	0 98	1 00	1 02	1 05	1 04
Ethyl ether	0 90	0 96	1 00	1 03	1 06	1 08
Ethyl iodide	0 80	0 90	1 00	1 08	1 13	1 14
Heptane	0 95	0 98	1 00	1 03	1 05	1 07
Hexane	0 89	0 94	1 00	1 03	1 07	1 09
Methyl alcohol (100%)	0 85	0 91	1 00	1 06	1 10	1 14
Methyl alcohol (60%)	0 74	0 89	1 00	1 09	1 17	1 21
Octane (n)	0 92	0 96	1 00	1 04	1 08	1 11
Pentane (n)	0 96	0 98	1 00	1 02	1 05	1 07
Sulfur dioxide	1 04	1 02	1 00	0 99	0 98	0 97
Sulfuric acid (98%)		0 76	1 00	1 45		
Sulfuric acid (60%)		0 90	1 00	1 17	1 30	1 33
Toluene	0 00	0 95	1 00	1 04	1 07	1 08
Water		0 77	1 00	1 15	1 26	1 36

[a] Average temperature of the film = (temperature of the tube wall + temperature of the main body of the liquid) ÷ 2

[b] Value in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 27.—DIAMETER-CORRECTION FACTOR FOR CASE 11

Outside Tube Dia, In	F_d
0 250	1 41
0 375	1 28
0 500	1 19
0 625	1 13
0 750	1 08
1 00	1 00
1 25	0 93
1 50	0 90
2 00	0 84
2 50	0 80
3 00	0 76
3 50	0 73
4 00	0 71
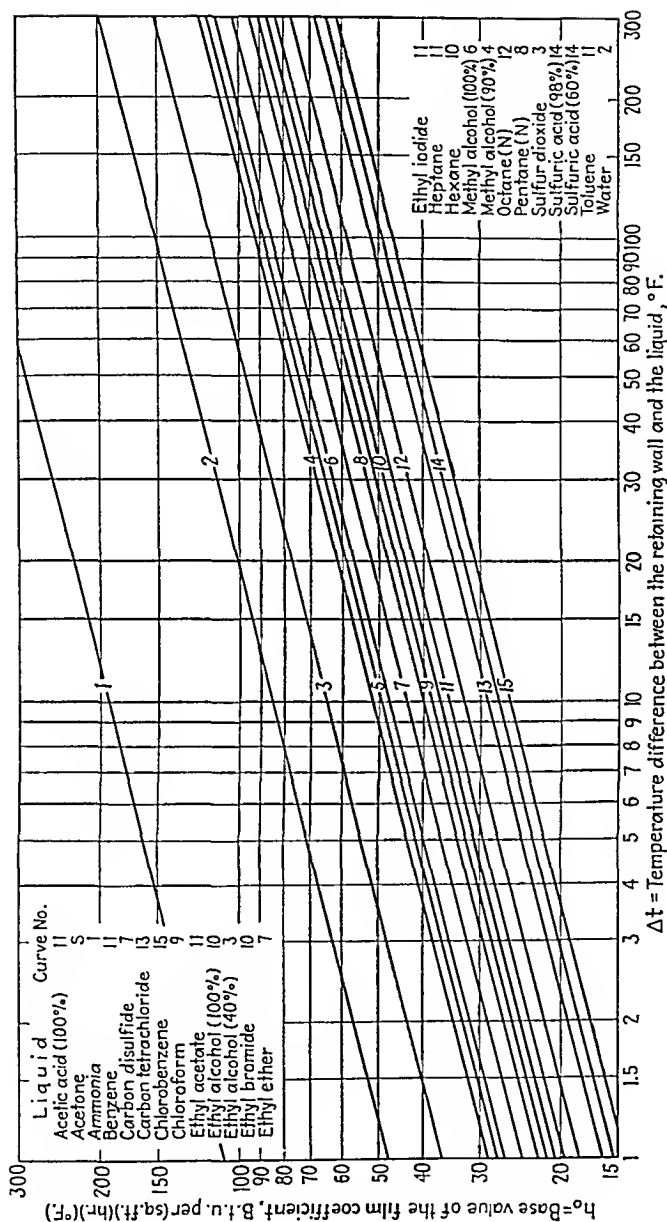
Fig. 31.—Base value of the film coefficient for Case 11.

Case 12 *Gases Heated outside Single Horizontal Tubes, Natural Convection* —The film coefficient can be determined by the equation

$$h = h_b \times F_t \times F_d \times F_p$$

where h = the film coefficient, $B.t.u.$/(sq ft)(hr)(°F), h_b = the base value of the film coefficient, from Fig 32, F_t = the temperature-correction factor, from Table 28, F_d = the diameter-correction factor, from Table 29, F_p = the pressure-correction factor, from Table 30

Note that the film coefficient h obtained from the foregoing equation is for heat transfer by *convection* only If the heat-transfer surface is exposed to other surfaces at different temperatures it will gain or lose heat by *radiation* also The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface, obtained from Table IV in the Appendix times the radiation coefficient h_r, obtained from Fig 10 in Chap II, times the convection coefficient h obtained from the equation above

TABLE 28—TEMPERATURE-CORRECTION FACTOR FOR CASE 12

Gas	Ave temp of film*						
	−100°F	0°F	100°F	200°F	300°F	400°F	500°F
	F t						
Acetone			1 00	1 12	1 27	1 41	1 5o
Acetylene	0 87	0 94	1 00	1 18	1 18	1 18	1 25
Air	1 03	1 01	1 00	0 92	0 97	0 95	0 23
Ammonia	0 87	0 91	1 00	1 0a	1 10	1 18	1 19
Benzene			1 00	1 17	1 36	1 38	1 30
Butane		0 92	1 00	1 06	1 16	1 21	1 31
Carbon oxide	0 88	0 9a	1 00	1 03	1 0a	1 07	1 00
Carbon monoxide	1 00	1 00	1 00	0 9a	0 96	0 97	0 96
Chlorine		0 01	1 00	0 99	0 99	0 98	0 97
Chloroform			1 00	1 07	1 14	1 20	1 27
Ethane	0 80	0 90	1 00	1 10	1 19	1 28	1 37
Ethyl acetate			1 00	1 15	1 25	1 43	1 57
Ethyl alcohol			1 00	0 95	0 87	0 96	0 82
Ethyl chloride			1 00	1 00	1 12	1 17	1 22
Ethylene	0 83	0 91	1 00	1 09	1 17	1 27	1 33
Ethyl ether			1 00	1 11	1 21	1 32	1 43
Helium	1 07	1 03	1 00	0 98	0 97		
Hydrogen	1 04	1 02	1 00	0 98	0 96	0 94	0 22
Hydrogen sulfide		0 99	1 00	1 01	1 01		
Methane	0 83	0 91	1 00	1 05	1 10	1 14	1 18
Methyl chloride		0 96	1 00	1 12	1 23	1 34	1 45
Nitric oxide	1 01	1 01	1 00	0 99	0 97	0 9a	0 95
Nitrogen	1 03	1 02	1 00	0 98	0 96	0 9a	0 91
Nitrous oxide	1 13	1 08	1 00	0 93	0 8a	0 81	0 87
Oxygen	1 02	1 01	1 00	0 99	0 97	0 96	0 93
Pentane (so)			1 00	1 11	1 14	1 18	1 23
Steam		0 99	1 00	1 02	1 04	1 06	1 08
Sulfur di ox			1 00	1 02	1 03	1 04	1 04

* Average temperature of the film = (temperature of the tube wall + temperature of the gas) ÷ 2
† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 29—DIAMETER-CORRECTION FACTOR FOR CASE 12

Outside tube dia in	F_d	Outside tube d a in	F_d
0 250	1 41	1 50	0 90
0 375	1 28	2 00	0 84
0 500	1 19	2 50	0 80
0 625	1 13	3 00	0 76
0 750	1 08	3 50	0 73
1 00	1 00	4 00	0 71
1 25	0 93		

TABLE 30—PRESSURE-CORRECTION FACTOR FOR CASE 12

Pressure lb per sq in gauge	F_p	Pressure lb per sq in gauge	F_p
0	1 00	100	2 79
10	1 39	150	3 3
20	1 54	200	3 8
30	1 74	250	4 2
40	1 93	300	4 6
50	2 10	350	5 0
60	2 26	400	5 3
70	2 40	450	5 6
80	2 54	500	5 9
90	2 67		
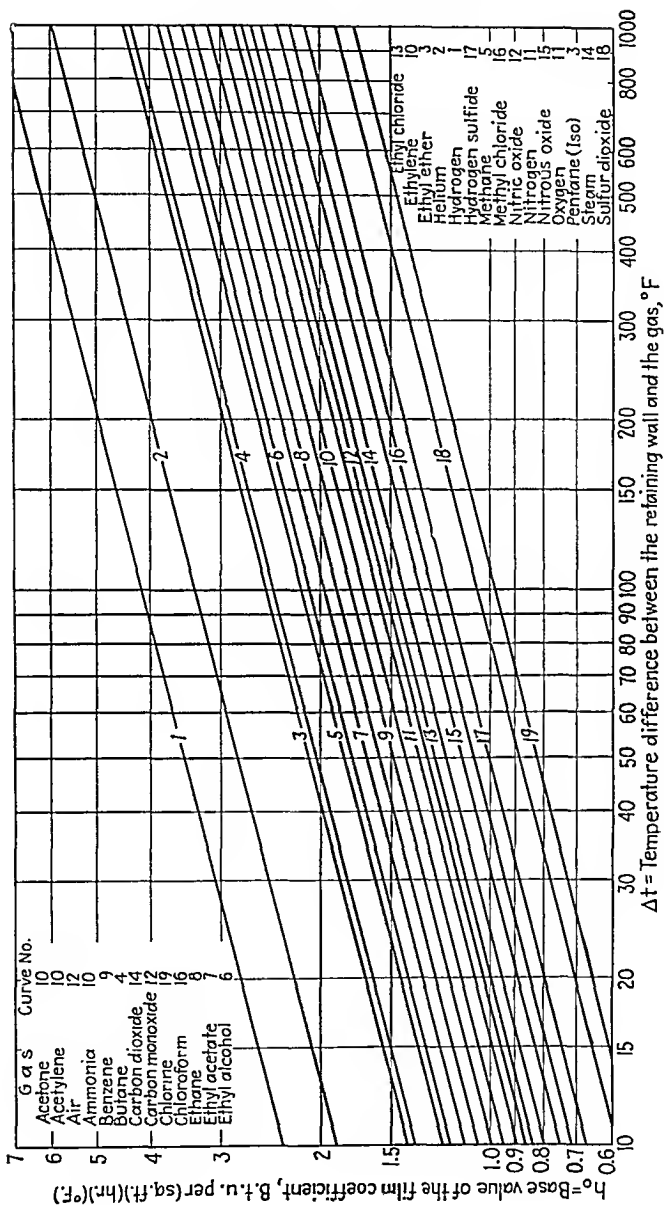
FIG. 32.—Base value of the film coefficient for Case 12.

Gas	Curve No.
Acetone	10
Acetylene	10
Air	12
Ammonia	10
Benzene	9
Butane	4
Carbon dioxide	14
Carbon monoxide	12
Chlorine	19
Chloroform	16
Ethane	8
Ethyl acetate	7
Ethyl alcohol	6

Ethyl chloride — 13
Ethylene — 10
Ethyl ether — 3
Helium — 2
Hydrogen — 1
Hydrogen sulfide — 17
Methane — 5
Methyl chloride — 16
Nitric oxide — 12
Nitrogen — 11
Nitrous oxide — 15
Oxygen — 11
Pentane (iso) — 3
Steam — 14
Sulfur dioxide — 18

Δt = Temperature difference between the retaining wall and the gas, °F

h_o = Base value of the film coefficient, B.t.u. per (sq.ft.)(hr.)(°F.)

Case 13   Liquids Heated inside or outside Vertical Tubes or on Vertical Plates, Low Velocities or Natural Convection Only —The film coefficient can be determined by the equation

$$h = h_b \times F_t$$

where h = the film coefficient B t u /(sq ft)(hr)(°F)
h_b = the base value of the film coefficient from Fig 33
F_t = the temperature-correction factor from Table 31

TABLE 31—TEMPERATURE-CORRECTION FACTOR FOR CASE 13

Liquid	Ave temp of the film*					
	0°F	50°F	100°F	150°F	200°F	250°F
	F_t†					
Acetic acid (100%)			1 00	0 99	0 89	0 93
Acetone	0 87	0 94	1 00	1 03	1 0o	1 0~
Ammonia	0 73	0 86	1 00	1 15	1 3°	1 51
Benzene		0 88	1 00	1 11	1 22	1 33
Carbon disulfide	0 91	0 97	1 00	1 03	1 00	1 09
Carbon tetrachloride	0 83	0 92	1 00	1 03	1 0o	1 0~
Chlorobenzene	1 00	1 00	1 00	0 99	0 99	0 98
Chloroform	0 8o	0 93	1 00	1 0~	1 13	1 20
Ethyl acetate	1 0	1 01	1 00	0 96	0 92	0 87
Ethyl alcohol (100%)	0 68	0 82	1 00	1 17	1 33	1 50
Ethyl alcohol (40%)	0 40	0 68	1 00	1 3	1 77	
Ethyl bromide	0 91	0 96	1 00	1 03	1 0o	1 0
Ethyl ether	0 87	0 94	1 00	1 0o	1 08	1 10
Ethyl iodide	0 1	0 87	1 00	1 12	1 23	1 3o
Heptane	0 89	0 94	1 00	1 0C	1 11	1 1
Hexane	0 88	0 94	1 00	1 06	1 09	1 12
Methyl alcohol (100%)	0 ~6	0 89	1 00	1 09	1 18	1 2o
Methyl alcohol (90%)	0 6	0 86	1 00	1 12	1 23	1 3;
Octane (n)	0 8o	0 94	1 00	1 07	1 13	1 20
Pentane (n)	0 91	0 96	1 00	1 03	1 0o	1 0~
Sulfur dioxide	1 00	1 00	1 00	1 00	1 00	1 00
Sulfuric acid (98%)		0 76	1 00	1 23	1 46	1 69
Sulfuric acid (60%)		0 76	1 00	1 19	1 31	1 57
Toluene	0 8o	0 92	1 00	1 07	1 15	1 22
Water		0 73	1 00	1 19	1 33	1 50

* Average temperature of the film = (temperature of the retaining wall + temperature of the main body of the liquid) ÷ 2
† Values in italics are based on extrapolated values of the physical properties of the fluids

Fig. 33.—Base value of the film coefficient for Case 13.

Liquid	Curve No.
Acetic acid (100%)	12
Acetone *	4
Ammonia *	17
Benzene	11
Carbon disulfide	7
Carbon tetrachloride	14
Chlorobenzene	15
Chloroform	9
Ethyl acetate	10
Ethyl alcohol (100%)	12
Ethyl alcohol (40%)	3
Ethyl bromide	9

Ethyl ether	5
Ethyl iodide	10
Heptane	11
Hexane	10
Methyl alcohol (100%)	6
Methyl alcohol (90%)	5
Octane (N)	13
Pentane (N)	8
Sulfur dioxide	1
Sulfuric acid (98%)	18
Sulfuric acid (60%)	16
Toluene	12
Water	2

* Multiply value of h_0 by 10

Δt = Temperature difference between the retaining wall and the liquid, °F

h_0 = Base value of the film coefficient, B.t.u. per (sq.ft.)(hr.)(°F.)

Case 14   Gases Heated inside or outside Vertical Tubes or on Vertical Plates, Natural Convection —The film coefficient can be determined by the equation

$$h = h_b \times F_t \times F_p$$

where h = the film coefficient, B t u /(sq ft)(hr)(°F)
h_b = the base value of the film coefficient, from Fig 34
F_t = the temperature-correction factor, from Table 32
F_p = the pressure-correction factor, from Table 33

Note that the film coefficient h obtained from the foregoing equation is for heat transfer by *convection* only   If the heat-transfer surface is exposed to other surfaces at different temperatures, it will gain or lose heat by *radiation* also   The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface, obtained from Table IV in the Appendix, times the radiation coefficient h_r, obtained from Fig 10 in Chap II, onto the convection coefficient h obtained from the equation above

TABLE 32—TEMPERATURE-CORRECTION FACTOR FOR CASE 14

Gas	Ave temp of the film*						
	−100°F	0°F	100°F	200°F	300°F	400°F	500°F
				F t†			
Acetone			1 00	1 06	1 13	1 21	1 28
Acetylene	1 07	1 04	1 00	0 97	0 93	0 89	0 87
Air	1 15	1 07	1 00	0 95	0 90	0 85	0 82
Ammonia	1 00	1 00	1 00	0 99	0 98	0 97	0 95
Benzene			1 00	1 00	1 19	1 33	1 46
Butane		0 89	1 00	1 01	1 03	1 01	1 00
Carbon dioxide	1 04	1 02	1 00	0 98	0 95	0 93	0 91
Carbon monoxide	1 14	1 04	1 00	0 94	0 89	0 85	0 81
Chlorine		1 07	1 00	0 95	0 90	0 87	0 84
Chloroform			1 00	1 01	1 02	1 04	1 05
Ethane	0 93	0 96	1 00	1 06	1 07		1 14
Ethyl acetate			1 00	1 08	1 15	1 22	1 27
Ethyl alcohol			1 00	0 94	0 88	0 82	0 76
Ethyl chloride			1 00	0 96	0 97	0 95	0 94
Ethylene	0 95	0 98	1 00	1 02	1 04	1 07	1 10
Ethyl ether			1 00	1 03	1 08	1 14	1 20
Helium	1 21	1 08	1 00	0 8o	0 90		
Hydrogen	1 18	1 05	1 00	0 93	0 88	0 82	0 78
Hydrogen sulfide		1 03	1 00	0 9o	0 89		
Methane	1 05	1 02	1 00	0 97	0 9o	0 93	0 91
Methyl chloride		0 01	1 00	1 07	1 13	1 18	1 23
Nitric oxide	1 15	1 07	1 00	0 04	0 88	0 83	0 79
Nitrogen	1 17	1 07	1 00	0 03	0 87	0 82	0 77
Nitrous oxide	1 26	1 11	1 00	0 91	0 84	0 77	
Oxygen	1 15	1 07	1 00	0 04	0 88	0 83	0 79
Pentane (iso)		0 08	1 00	1 01	1 11	1 18	1 22
Steam			1 00	0 98	0 93	0 91	0 89
Sulfur dioxide			1 00	0 96	0 93	0 90	0 88

* Average temperature of the film = (temperature of the retaining wall + temperature of the main body of the gas) ÷ 2
† Values in italics are based on extrapolated values of the physical properties of the fluids

TABLE 33—PRESSURE-CORRECTION FACTOR FOR CASE 14

Press in lb per sq in gauge	F_p	Pressure lb per sq in gauge	F_p
0	1 00	100	3 93
10	1 41	150	5 0
20	1 77	200	6 0
30	2 10	250	6 9
40	2 40	300	7 7
50	2 65	350	8 5
60	2 9?	400	9 3
70	3 21	450	10 0
80	3 48	500	10 7
90	3 70		

Gas	Curve No.
Acetone	8
Acetylene	11
Air	15
Ammonia	13
Benzene	6
Butane	2
Carbon dioxide	14
Carbon monoxide	6
Chlorine	18
Chloroform	12
Ethane	9
Ethyl acetate	4
Ethyl alcohol	5
Ethyl chloride	10

Gas	Curve No.
Ethylene	11
Ethyl ether	1
Helium	3
Hydrogen	17
Hydrogen sulfide	16
Methane	15
Methyl chloride	16
Nitric oxide	15
Nitrogen	16
Nitrous oxide	14
Oxygen	1
Pentane (isc)	16
Steam	16
Sulfur dioxide	17

h_o = Base value of the film coefficient, B.t.u. per (sq.ft.)(hr.)(°F.)

Δt = Temperature difference between the retaining wall and the gas, °F.

Fig. 34.—Base value of the film coefficient for Case 14.

Case 15   Liquids Heated or Cooled inside Coils Turbulent Flow — Approximate values of the film coefficient can be determined by the equation

$$h_{coils} = 1\ 2 \times h_{ubes}$$

where h_{tubes} for *heating* is obtained from Case 1 and h_{tubes} for *cooling* is obtained from Case 2

Case 16   Gases Heated or Cooled inside Coils, Turbulent Flow — The film coefficient can be determined by the equation

$$h_{cor\ ls} = \left[1 + 3\ 54\left(\frac{d}{d_c}\right) \right] \times h_{tubes}$$

where d = the inside diameter of the pipe or tube in

d_c = the diameter of the coil in

h_{tubes} = the film coefficient for gases inside straight tubes obtained from Case 3 or 4   B t u /(sq ft)(hr)(°F)

Case 17   Liquids or Gases Heated or Cooled outside Coils, Natural or Forced Convection — The film coefficients for liquids or gases heated or cooled *outside coils* are approximately equal to the film coefficients for liquids or gases heated or cooled *outside single tubes* and can be obtained from Case 5 6 7 11 or 12

Case 18   Air Heated on Horizontal Plates, Natural Convection — The film coefficient for air heated by natural convection on large flat plates (3 sq ft or more) can be determined by the equations

$$h_H = 1\ 27 \times h_V \text{ (plate facing upward)}$$

or

$$h_H = 0\ 67 \times h_V \text{ (plate facing downward)}$$

where h_H = the film coefficient for horizontal plates B t u /(sq ft)(hr)(°F)

h_V = the film coefficient for vertical plates obtained from Case 14 B t u /(sq ft)(hr)(°F)

Note that the film coefficients obtained from the foregoing equations are for heat transfer by *convection* only   If the heat-transfer surface is exposed to other surfaces at different temperatures it will gain or lose heat by *radiation* also   The film coefficient for the combined effect of convection and radiation can be obtained by adding the product of the emissivity of the surface obtained from Table IV in the Appendix times the radiation coefficient h obtained from Fig 10 in Chap 11 onto the convection coefficient h obtained from the equation above

Case 19. Petroleum Oils Heated inside Horizontal or Vertical Tubes, Streamline Flow.—The film coefficient can be determined by the equation

$$h = h_0 \times F_t \times F_d,$$

where h = the film coefficient, based on the arithmetic mean of the terminal temperature differences, B.t.u./(sq. ft.)(hr.)(°F.); h_0 = the base value of the film coefficient, from Fig. 35; F_t = the temperature-correction factor, from Table 34; F_d = the diameter-correction factor, from Table 35.

TABLE 34.—TEMPERATURE-CORRECTION FACTOR FOR CASE 19

Ave. temp. of oil, °F.	F_t
100	1.00
200	1.00
300	0.99
400	0.98
500	0.97

TABLE 35.—DIAMETER-CORRECTION FACTOR FOR CASE 19

Inside tube dia., in.	F_d	Inside tube dia., in.	F_d
0.2	5.00	1.2	0.83
0.3	3.33	1.3	0.77
0.4	2.50	1.4	0.71
0.5	2.00	1.5	0.67
0.6	1.67		
0.7	1.43	2.0	0.50
0.8	1.25	2.5	0.40
0.9	1.11	3.0	0.33
1.0	1.00	3.5	0.27
1.1	0.91	4.0	0.25

TABLE 36.—MAXIMUM PERMISSIBLE VALUES OF W'/d FOR CASE 19
W' = rate of flow per tube, lb. per hr. per tube; d = inside tube dia., in.

Viscosity, Saybolt Universal seconds	Gravity, °A.P.I.					
	10°	20°	30°	40°	50°	60°
	$\left(\dfrac{W'}{d}\right)_{max}$					
40	1,300	1,400	1,500	1,600	1,700	1,800
50	2,300	2,500	2,700	2,800	3,000	3,200
60	3,200	3,500	3,700	3,900	4,100	4,400
80	4,800	5,200	5,500	5,800	6,200	6,500
100	6,400	6,800	7,300	7,800	8,200	8,700
150	10,100	10,900	11,600	12,300	13,000	13,700
200	13,800	14,800	15,700	16,700	17,700	18,700
300	20,600	22,100	23,600	25,000	26,400	27,900
400	27,600	29,600	31,500	33,400	35,400	37,400
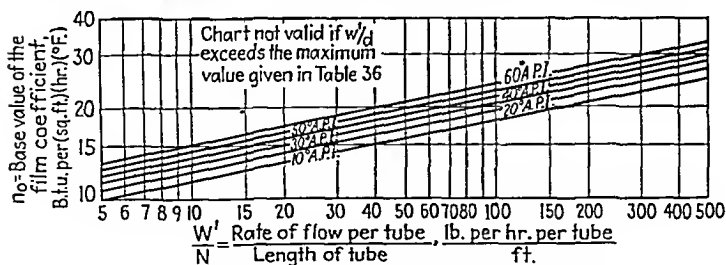
FIG. 35.—Base value of the film coefficient for Case 19.

Case 20. Petroleum Oils Cooled inside Horizontal or Vertical Tubes, Streamline Flow.—Approximate values of the film coefficient can be determined by the equation

$$h_C = 0.7 \times h_H,$$

where h_C = the film coefficient for *cooling*, B.t.u./(sq. ft.)(hr.)(°F.); h_H = the film coefficient for *heating*, obtained from Case 19, B.t.u./(sq. ft.)(hr.)(°F.).
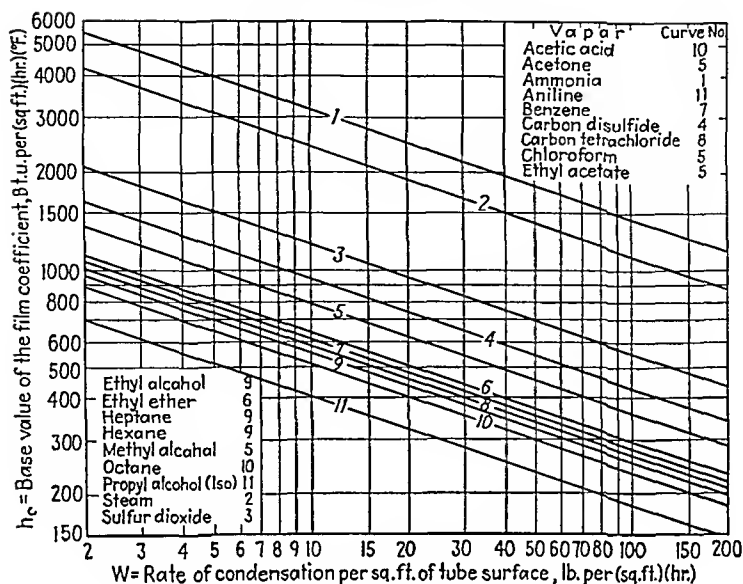
Fig. 36.—Base value of the film coefficient for Cases 21 and 22b.

Case 22   Condensation of Pure Saturated Vapors on Vertical Tubes or Plates—The method of evaluating the film coefficient for this case depends upon whether $W \times N$ is greater or less than the critical value in Table 40

TABLE 40—CRITICAL VALUE OF THE PRODUCT $W \times N$ FOR CASE 22

W = rate of condensation per square foot of heat-transfer surface, lb /(sq ft)(hr)

N = height of the surface, ft

Vapor	Temp of condensate film*					
	60°F	100°F	150°F	200°F	250°F	300°F
	$(W \times N)_{\text{critical}}$					
Acetic acid	1150	770	550	390	270	200
Acetone	290	220	170	140	110	90
Ammonia	60	50	30	20	10	10
Aniline		2300	1100	620	370	220
Benzene	580	410	290	210	160	120
Carbon disulfide	310	260	220	190	170	140
Carbon tetrachloride	900	630	450	310	220	170
Chloroform	500	380	290	230	190	160
Ethyl acetate	420	300	220	170	140	110
Ethyl alcohol	1080	690	430	260	190	130
Ethyl ether	200	160	120	100	80	70
Heptane	300	270	200	160	130	110
Hexene	270	210	170	130	110	90
Methyl alcohol	540	360	260	190	140	110
Octane	500	360	270	210	160	130
Propyl alcohol (iso)		1080	550	340	230	170
Steam	1070	540	340	230	160	150
Sulfur dioxide	290	230	190	150	120	100

*May be assumed equal to the temperature of the tube wall

a If $W \times N$ is greater than the critical value given in Table 40

$$h = h_e \times F_t \times F_{N'}$$

where h = the film coefficient B t u /(sq ft)(hr)(°F)

h_e = the basic value of the film coefficient, from Fig 37

F_t = the temperature-correction factor, from Table 41

$F_{N'}$ = the correction factor for the height of the surface from Table 42

TABLE 41—TEMPERATURE-CORRECTION FACTOR FOR CASE 22*a*

Vapor	Temp of condensate film*					
	50°F	100°F	150°F	200°F	250°F	300°F
	F_t†					
Acetic acid		1 00	1 22	1 49	1 81	2 19
Acetone	0 81	1 00	1 29	1 41	1 61	1 82
Ammonia	0 00	1 00	1 56	2 38		
Aniline		1 00	2 89	4 20	7 70	
Benzene	0 69	1 00	1 40	1 85	2 53	3 40
Carbon disulfide	0 87	1 00	1 12	1 23	1 33	1 43
Carbon tetrachloride	0 71	1 00	1 23	1 59		
Chloroform	0 77	1 00	1 29	1 62		
Ethyl acetate	0 71	1 00	1 33	1 71	2 16	2 6o
Ethyl alcohol	0 54	1 00	1 58	2 34	3 50	4 95
Ethyl ether	0 79	1 00	1 22	1 46	1 71	1 98
Heptane	0 73	1 00	1 30	1 63	1 98	2 36
Hexane	0 78	1 00	1 24	1 49	1 75	2 00
Methyl alcohol	0 69	1 00	1 44	1 97	2 60	3 30
Octane	0 75	1 00	1 29	1 63	2 00	2 40
Propyl alcohol (iso)		1 00	2 00	3 16	4 47	
Steam	0 46	1 00	1 69	2 52	3 75	4 9o
Sulfur dioxide	0 89	1 00	1 07	1 11	1 14	1 16

*May be assumed equal to the temperature of the wall
†Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 42.—CORRECTION FACTOR FOR THE HEIGHT OF THE SURFACE, CASE 22a

Height of surface, ft.	F_N'	Height of surface, ft.	F_N'
2	0 53	12	1 08
4	0 69	14	1 15
6	0 82	16	1 21
8	0 91	18	1 27
10	1 00	20	1 32
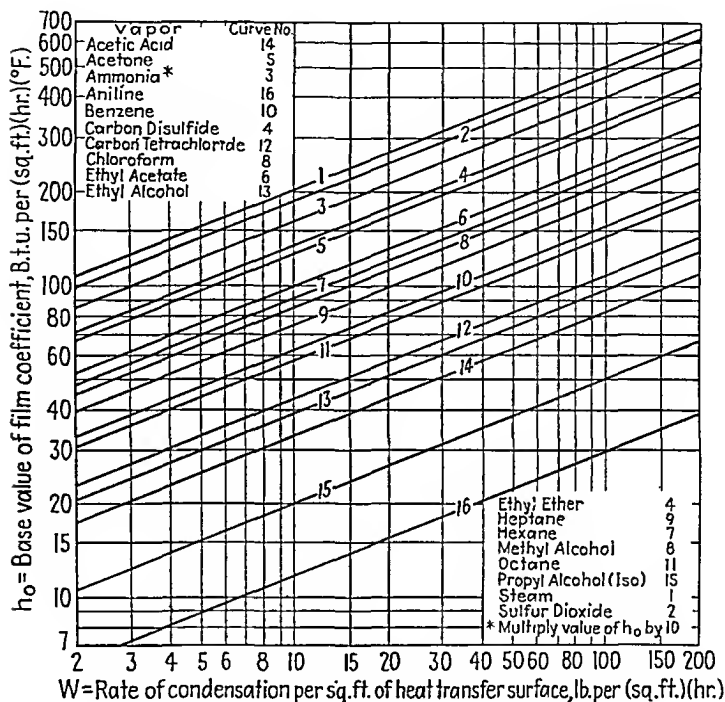
FIG. 37.—Base value of the film coefficient for Case 22a.

b. If $W \times N$ is less than the critical value given in Table 40,

$$h = 0.29 \times h_0 \times F_t \times F_N'',$$

where h = the film coefficient, B.t.u./(sq. ft.)(hr.)(°F.).

h_0 = the base value of the film coefficient, from Fig. 36.

F_t = the temperature-correction factor, from Table 37.

F_N'' = the correction factor for the height of the surface, from Table 43.

TABLE 43.—CORRECTION FACTOR FOR THE HEIGHT OF THE SURFACE, CASE 22b

Height of surface, ft.	F_N''	Height of surface, ft.	F_N''
2	1 71	12	0 94
4	1 36	14	0 89
6	1 19	16	0 85
8	1 08	18	0 82
10	1 00	20	0 79

Case 23   Liquids Boiling on Horizontal or Vertical Plates —Conservative values of the film coefficient for liquids boiling at atmospheric pressure on horizontal or vertical plates can be obtained from Fig 38   At values of Δt below the range given in this figure the film coefficient for boiling is approximately equal to the film coefficient for free convection on vertical surfaces and can be obtained from Case 13

The film coefficient for *water* boiling at pressures other than atmospheric can be obtained by multiplying the value obtained from Fig 38 by the pressure-correction factor given in Table 44

TABLE 44—PRESSURE-CORRECTION FACTOR FOR WATER.

Absolute Pressure Atm *	F_p
0 2	0 62
0 4	0 78
0 6	0 88
0 8	0 94
1 0	1 00
2 0	1 16
4 0	1 32
6 0	1 40
8 0	1 46
10 0	1 51
15 0	1 60

* 1 atm = 14 7 lb per sq in

Case 24   Liquids Boiling inside Tubes —If the motion of the liquid is the result of natural convection only conservative values of the film coefficient can be determined by the equation

$$h_T = 1\ 25 \times h_P$$

where h_T = the film coefficient for liquids boiling inside tubes B t u /(sq ft)(hr)(°F)

h_P = the film coefficient for liquids boiling on flat plates obtained from Case 23 B t u /(sq ft)(hr)(°F)

If the liquid is forced through the tubes at high velocities by means of a pump the film coefficient is approximately the same as it would be if no evaporation took place and can be obtained from Case 1
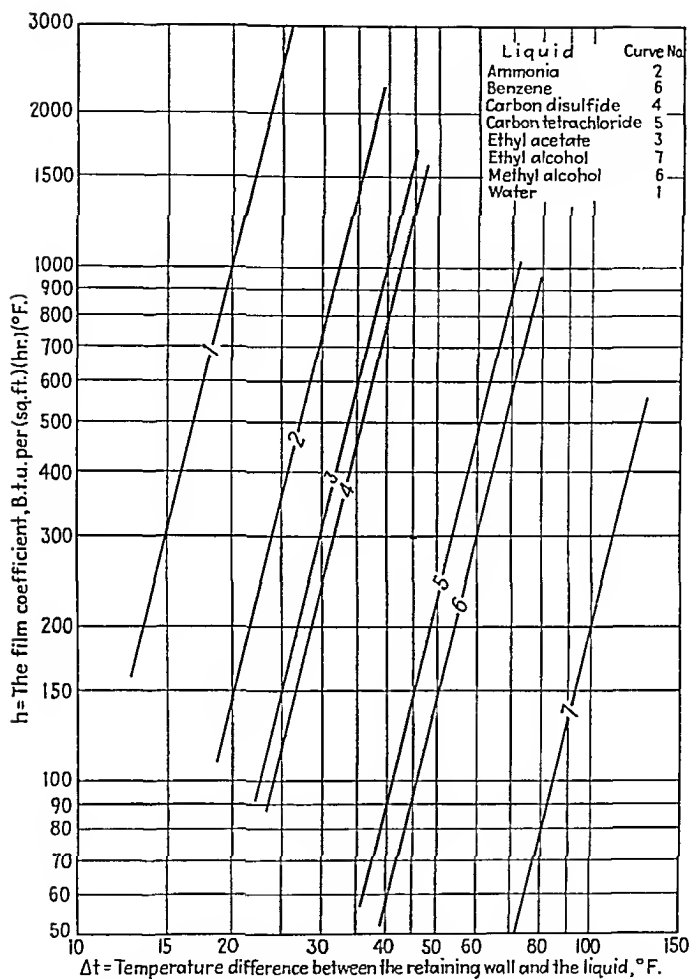
FIG. 38.—Film coefficients for liquids boiling at atmospheric pressure on horizontal or vertical plates, Case 23.

Case 25   Scale Coefficients.—The scale coefficient h_s depends upon the thickness and the thermal conductivity of the scale   If the thickness of the scale can be estimated, rough values of the scale coefficient can be calculated by the equation

$$h_s = \frac{k_s}{L_s}$$

where h_s = the scale coefficient   B t u /(sq ft)(hr)(°F)

   k_s = the apparent thermal conductivity of the scale, B t u /(ft)(hr)(°F)

   L_s = the thickness of the scale, ft

Although the apparent thermal conductivity of scale varies with the kind and with the porosity of the scale, a mean value of 1 0 B t u /(ft)(hr)(°F) may be used for k_s

The following values of h_s may be used for the kinds of convection indicated

Kind of Convection	h_s B t u /(sq ft)(hr)(°F)
Condensing steam	3000
Condensing petroleum vapors	500
Heating muddy water	600
Heating residual petroleum stocks	200

CHAPTER V

CALCULATION OF PRESSURE DROP

33. Introduction.—High fluid velocities in heat-transfer equipment are desirable because the rate of heat transfer by convection is higher at high than at low velocities. The pressure drop through the equipment is also higher at high velocities, however, and therefore the maximum velocity that can be used is fixed by the permissible pressure drop.

The probable over-all pressure drop through any proposed equipment is equal to the sum of the individual pressure drops that the fluid will undergo. For example, the total pressure drop that the fluid flowing inside the tubes of a heat exchanger will undergo is equal to (1) the pressure drop caused by the abrupt contraction in cross section where the fluid enters the tubes plus (2) the pressure drop caused by friction in the tubes minus (3) the pressure rise caused by the abrupt enlargement in cross section where the fluid leaves the tubes and, if the exchanger is multipass, plus (4) the pressure drop caused by each reversal in the direction of flow. If the fluid is a gas, there will be an additional change in pressure caused by changes in the density of the gas as it is heated or cooled. Thus, changes in density will cause the gas to accelerate or slow down as it flows through the tubes, which will be accompanied by changes in pressure. Charts and tables which can be used to evaluate readily each of these individual pressure drops for various fluids are given at the end of this chapter.

34. Calculation of the Charts and Tables.—The equations upon which the charts and tables are based are given in this section. The symbols used in these equations are defined in Table 45.

TABLE 45.—NOMENCLATURE

A Inside cross-sectional area, sq. ft.
D Inside tube or pipe diameter, ft.
d Inside tube or pipe diameter, in.
f Friction factor, no units.
ΔF Energy decrease, ft.-lb. per lb.

TABLE 45—Nomenclature —(*Continued*)

G	Weight velocity, lb /(sec)(sq ft of cross section)
g	Acceleration of gravity, 322 ft per sec ²
K	Contraction coefficient no units
N	Length of tube or pipe ft
P	Pressure, lb per sq ft abs
ΔP	Pressure rise or drop, lb per sq ft abs
R	Gas constant, ft per °F
T	Temperature, °F abs
t	Temperature, °F
t_w	Tube-wall temperature, °F
V	Linear velocity, ft per sec
v	Specific volume, cu ft per lb
W'	Rate of flow, lb per hr per tube
μ	Absolute viscosity, lb /(ft)(sec)
ρ	Density, lb per cu ft

CASE 1 LIQUIDS, PRESSURE DROP INSIDE SMOOTH TUBES, TURBULENT FLOW —For isothermal, turbulent flow of fluids inside straight, smooth tubes, the pressure drop per foot of length can be calculated by the Fanning equation

$$\frac{\Delta P}{N} = \frac{4f\rho V^2}{2gD} \qquad (48)$$

The friction factor f has been determined experimentally by numerous investigators, and for the range $DV\rho/\mu$ equal to 4,000 to 1,000 000, their results can be represented approximately by the equation

$$f = \frac{0 0653}{\left(\frac{DV\rho}{\mu}\right)^{0\ 275}} \qquad (49)$$

This equation is valid only for isothermal flow , but, as shown by McAdams,[1] it can also be used as an approximation for the heating or cooling of any *liquid* if the viscosity μ is evaluated at the arithmetic average of the tube-wall temperature and the temperature of the main body of the liquid   Figure 39 and Tables 47 end 48 are based on Eqs (48) and (49), and the values of $V \times d$ corresponding to $DV\rho/\mu$ equal to 4,000 are given in Table 49

CASE 2 PETROLEUM OILS, PRESSURE DROP INSIDE TUBES OR PIPES, STREAMLINE FLOW —For streamline flow of petroleum oils inside straight tubes or pipes, McAdams[1] recommends that

---

[1] McAdams W H ' Heat Transmission," p 112 McGraw-Hill Book Company Inc 1933

the pressure drop per foot of length be calculated by the equation

$$\frac{\Delta P}{N} = \frac{35.2\mu' V}{gD^2}, \tag{50}$$

where μ' is evaluated at the average temperature t' defined by the equation

$$t' = \frac{3t + t_w}{4}. \tag{51}$$

In the latter equation, t represents the temperature of the main body of the oil and t_w represents the temperature of the tube wall. Equation (50) is valid only if the value of $DV\rho/\mu$ is less than 2,100. Figure 40 and Table 50 are based on Eq. (50) and on the equation

$$W'' = \frac{\pi D^2}{4} \times V \times 3,600\rho. \tag{52}$$

The values of W''/d corresponding to $DV\rho/\mu$ equal to 2,100 are given in Table 51.

CASE 3. LIQUIDS, PRESSURE DROP AT THE ENTRANCE TO TUBES.—The pressure drop caused by an abrupt contraction in cross section, such as occurs at the entrance to unflared tubes, can be calculated by the equation

$$\Delta P = \left[\frac{(V_2^2 - V_1^2)}{2g} + K \left(\frac{V_2^2}{2g} \right) \right] \times \rho, \tag{53}$$

where V_1 is the average velocity in the larger section and V_2 is the average velocity in the smaller. The first term in the

TABLE 46.—CONTRACTION COEFFICIENT FOR SHARP-EDGED ENTRANCES*

Ratio of Larger to Smaller Area	K
1 0	0 00
1 5	0 13
2 0	0 19
2 5	0 24
3 0	0.27
3 5	0 30
4.0	0.32
6.0	0.38
8 0	0.41
10 0	0 43

* Taken from H. J. Hughes and A. T. Safford, "Hydraulics," The Macmillan Company, 1926.

right-hand member of this equation represents the pressure drop required to increase the *velocity* of the fluid, and the *second* represents the pressure drop caused by friction. Values of the contraction coefficient K for sharp-edged entrances are given in Table 46. Figure 41 and Table 52 are based on Eq (53), on these values of K, and on the equation

$$A_1 V_1 = A_2 V_2 \qquad (54)$$

For rounded or conical entrances, the value of K is quite small, and the pressure drop caused by friction is practically negligible.

CASE 4   LIQUIDS, PRESSURE RISE AT THE OUTLET FROM TUBES —The rise in pressure caused by an abrupt enlargement in cross section, such as occurs at the outlet from tubes, can be calculated by the equation

$$\Delta P = \left[\frac{(V_1^2 - V_2^2)}{2g} - \frac{(V_1 - V_2)^2}{2g} \right] \times \rho, \qquad (55)$$

where V_1 is the average velocity in the smaller section and V_2 is the average velocity in the larger. The first term in the right-hand member of this equation represents the increase in pressure caused by the decrease in the velocity of the fluid, and the second represents the pressure drop caused by friction. It may be noted that the first term is always larger than the second and that ΔP therefore always represents a pressure rise. Figure 42 and Table 53 are based on Eqs (54) and (55).

CASE 5   GASES, PRESSURE DROP INSIDE SMOOTH TUBES, TURBULENT FLOW —Figure 43 and Tables 54 and 55 are based on Eqs (48) and (49) and on the assumption that the gases obey the perfect-gas law $Pv = RT$. Although Eq (49) is based on data for isothermal flow, it may also be used if the gases are being heated or cooled, because the viscosities of gases do not change rapidly with temperature.

CASE 6   GASES, PRESSURE DROP AT THE ENTRANCE TO TUBES —Figure 44 and Table 56 are also based on Eqs (53) and (54) and on the values of the contraction coefficient K given in Table 46. The use of Eq (54) involves the assumption that the density of the gas changes only slightly during the contraction. This assumption is permissible for moderate rates of flow. The values of the constant b given in Table 57 are based on the assumption that the gases obey the perfect-gas law $Pv = RT$

CASE 7. GASES, PRESSURE RISE AT THE OUTLET FROM TUBES.—Figure 45 and Table 58 are also based on Eqs. (54) and (55). In using Eq. (54), it is again assumed that the density of the gas changes only slightly during the process.

CASE 8. GASES, PRESSURE RISE OR DROP CAUSED BY CHANGES IN DENSITY.—If a gas is heated as it flows through a tube of constant cross section, it will leave the tube at a higher velocity than it enters, because its density will be decreased. This increase in velocity will be accompanied by a drop in pressure which can be determined as follows: The energy required to accelerate the gas is equal to the increase in the kinetic energy of the gas; *i.e.*,

$$\Delta F = \frac{V_2{}^2}{2g} - \frac{V_1{}^2}{2g},$$

where ΔF is measured in foot-pounds per pound of gas. Since the weight rate of flow $G = V_1/v_1 = V_2/v_2$, the foregoing equation can be written

$$\Delta F = \frac{G^2}{g}(v_2 - v_1)\left(\frac{v_2 + v_1}{2}\right)$$
$$= \frac{G^2}{g}(v_2 - v_1) \times v_{ave},$$

or

$$\Delta P = \frac{G^2}{g}(v_2 - v_1). \tag{56}$$

Equation (56) also applies if the gas is cooled, the negative value obtained indicating that the pressure rises.

Figure 46 is based on Eq. (56). This equation can also be applied to liquids; but since the densities of liquids vary only slightly with changes in temperature, the values of ΔP are negligible.

CASE 9. PRESSURE DROP CAUSED BY A REVERSAL IN THE DIRECTION OF FLOW.—The pressure drop caused by a reversal in the direction of flow depends upon the design of the passage in which the reversal takes place. This loss may be relatively large for liquids but is usually negligible for gases.

CASE 10. LIQUIDS OR GASES, PRESSURE DROP IN ANNULAR SPACES OR OUTSIDE TUBE BUNDLES, DIRECTION OF FLOW PARALLEL TO TUBES.—For *turbulent* flow, the pressure drop in annular spaces or outside tube bundles can be determined from

Note that this value does not include the loss of head due to differences in the elevation of the first and last headers

Illustrative Problem 2—A two-stage air compressor handles 500 cu ft of free air per minute (measured at 70°F and atmospheric pressure) The air is cooled between the stages from 240 to 90°F in an intercooler having the following specifications

$$
\begin{aligned}
&\text{Total number of tubes} = 40 \\
&\text{Length of tubes} = 8 \text{ ft} \\
&\text{Diameter of tubes} = \tfrac{5}{8} \text{ in o d} \\
&\text{Tube wall thickness} = \text{No 16 B W G}
\end{aligned}
$$

Ratio of cross sectional area of headers

to cross sectional area of tubes $= 2.5$

Determine the pressure drop through the intercooler if the average pressure of the air in the intercooler is 25 lb per sq in gauge

Solution—The density of the air entering the compressor (see Table 57) is $270 \times 147/(70 + 460) = 0.075$ lb per cu ft, and the cross-sectional area of each tube is 0.1921 sq in Therefore, the weight velocity in the tubes is

$$
G = \frac{500 \times 0.075}{60} \times \frac{144}{40 \times 0.1921}
$$

$$
= 117 \text{ lb /(sec)(sq ft)}
$$

The density of the air entering the intercooler is

$$
270 \times \frac{(25 + 147)}{(240 + 460)} = 0.153 \text{ lb per cu ft },
$$

and the density of the air leaving is $270 \times (25 + 147)/(90 + 460) = 0.202$ lb per cu ft The average density of the air in the intercooler is 0.178 lb per cu ft

The pressure drop caused by friction inside the tubes can be determined by Case 5 Thus from Fig 43 $(\Delta p/\Lambda)_5 = 0.048$ lb (sq in)/(ft), and from Tables 54 and 55 $F_t = 114$ (average air temperature $= 165°F$) and $F_d = 238$ (inside tube diameter $= 0.495$ in) The absolute pressure of the air is $(25 + 147)/147 = 270$ atm abs Therefore, since the tubes are 8 ft long

$$
\Delta p_1 = 0.048 \times 114 \times 238 \times \frac{1}{270} \times 8
$$

$$
= 0.39 \text{ lb per sq in}
$$

The pressure drop at the entrance to the tubes can be determined by Case 6 Thus, from Fig 44 $\Delta p_6 = 0.12$, and from Table 56, $F_v = 0.86$ Therefore,

$$
\Delta p_2 = 0.12 \times 0.86
$$

$$
= 0.10 \text{ lb per sq in}
$$

The pressure *rise* at the outlet from the tubes can be determined by

Case 7. Thus, from Fig. 45, $\Delta p_0 = 0.035$ lb. per sq. in.; and from Table 58, $F_r = 1.28$. Therefore,

$$\Delta p_3 = -0.035 \times 1.28$$
$$= -0.04 \text{ lb. per sq. in.}$$

The change in pressure caused by the increase in the density of the air can be determined by Case 8. Thus, from Fig. 46, $p_0' = 0.19$ lb. per sq. in. and $p_0'' = 0.14$ lb. per sq. in. Therefore,

$$\Delta p_4 = 0.14 - 0.19$$
$$= -0.05 \text{ lb. per sq. in.}$$

Finally, the total pressure drop through the intercooler is the sum of these individual drops:

$$\Delta p = 0.39 + 0.10 - 0.04 - 0.05$$
$$= 0.40 \text{ lb. per sq. in.}$$

Illustrative Problem 3.—A petroleum oil flows through the tubes of a heater at the rate of 1,500 lb. per hr. per tube. The tubes are 10 ft. long, $\frac{3}{4}$ in. o.d., have No. 18 B.W.G. walls (0.652 in. i.d.), and are at an average temperature of 200°F. Estimate the pressure drop through the tubes if the oil is heated from 60 to 160°F. and has the following properties:

$$\text{Gravity} = 20° \text{ A.P.I.}$$
$$\text{Viscosity at } 95°\text{F.} = 200 \text{ Saybolt Universal seconds.}$$
$$\text{Viscosity at } 160°\text{F.} = 60 \text{ Saybolt Universal seconds.}$$
$$\text{Viscosity at } 170°\text{F.} = 55 \text{ Saybolt Universal seconds.}$$

Solution.—The pressure drop can be determined by Case 2 since (W''/d) is less than the maximum permissible value given in Table 51. Thus, at 160°F., $(W''/d)_{max} = 3,500$, while $(W''/d) = 1,500/0.652 = 2,300$.

The "film" temperature of the oil on entering the tubes is

$$\frac{(3 \times 60 + 200)}{4} = 95°\text{F.,}$$

and the viscosity at this temperature is 200 Saybolt Universal seconds. Hence, from Fig. 40, $(\Delta p/N)_0 = 0.037$ lb./(sq. in.)/(ft.); and from Table 50, $F_d = 5.50$. Therefore, $(\Delta p/N) = 0.037 \times 5.50 = 0.204$ lb./(sq. in.)/(ft.).

Similarly, the film temperature of the oil on leaving the tubes is

$$\frac{(3 \times 160 + 200)}{4} = 170°\text{F.,}$$

the viscosity is 55 Saybolt Universal seconds, $(\Delta p/N)_0 = 0.0075$ lb./(sq. in.)/(ft.), and $(\Delta p/N) = 0.0075 \times 5.50 = 0.041$ lb./(sq. in.)/(ft.).

The average pressure drop per foot of length is somewhere between 0.041 and 0.204 lb./(sq. in.)/(ft.). Taking the arithmetic average of these two values, the total pressure drop is probably close to $10 \times 0.123 = 1.23$ lb. per sq. in.

Case 1. Liquids, Pressure Drop inside Smooth Tubes, Turbulent Flow.— The pressure drop per foot of length can be determined by the equation $(\Delta p/N) = (\Delta p/N)_b \times F_t \times F_d$, where $(\Delta p/N)$ = the pressure drop per foot of length, lb/(sq in.)(ft of length), $(\Delta p/N)_b$ = the base value of the pressure drop per foot of length, from Fig. 39, F_t = the temperature-correction factor, from Table 47, F_d = the diameter-correction factor, from Table 48.

TABLE 47—TEMPERATURE-CORRECTION FACTOR FOR CASE 1

Liquid	Density at 70°F lb per cu ft	Ave temp of the film F_t					
		0°F	50°F	100°F	150°F	200°F	250°F
Acetic acid (100%)	65 4			1 00	0 90	0 82	0 75
Acetic acid (50%)	66 0		1 20	1 00	0 87	0 77	0 69
Acetone	49 2	1 18	1 09	1 00	0 09	0 79	0 67
Ammonia	35 6	1 40	1 19	1 00	0 85	0 67	0 53
Amyl acetate	51 0	1 24	1 11	1 00	0 90	0 81	0 72
Amyl alcohol (iso)	50 7	1 40	1 21	1 00	0 85	0 74	0 65
Aniline	64 0		1 17	1 00	0 86	0 74	0 64
Benzene	54 8		1 12	1 00	0 81	0 81	0 76
Brine (CaCl₂) (25%)	76 7	1 31	1 13	1 00	0 89	0 79	0 69
Butyl alcohol (iso)	49 8	1 44	1 19	1 00	0 87	0 78	0 73
Carbon disulfide	78 7	1 14	1 07	1 00	0 94	0 88	0 82
Carbon tetrachloride	99 5	1 36	1 14	1 00	0 00	0 84	0 81
Chlorobenzene	73 0	1 27	1 13	1 00	0 89	0 81	0 76
Chloroform	92 7	1 21	1 10	1 00	0 91	0 83	0 76
Ethyl acetate	56 3	1 23	1 11	1 00	0 91	0 84	0 77
Ethyl alcohol (100%)	49 2	1 34	1 15	1 00	0 87	0 77	0 71
Ethyl alcohol (40%)	60 8	1 42	1 22	1 00	0 78	0 67	
Ethyl bromide	90 8	1 17	1 08	1 00	0 92	0 84	0 77
Ethylene glycol (50%)	66 5	1 49	1 22	1 00	0 82	0 69	0 61
Ethyl ether	44 0	1 22	1 10	1 00	0 92	0 83	0 78
Ethyl iodide	129 8	1 20	1 09	1 00	0 92	0 86	0 80
Glycerol (50%)	70 3	1 61	1 23	1 00	0 63	0 67	0 55
Heptane	42 6	1 23	1 10	1 00	0 01	0 83	0 75
Hexane	41 1	1 19	1 09	1 00	0 91	0 83	0 78
Methyl alcohol (100%)	49 4	1 24	1 12	1 00	0 89	0 76	0 64
Methyl alcohol (90%)	55 1	1 25	1 12	1 00	0 89	0 80	0 73
Methyl alcohol (40%)	56 2	1 40	1 23	1 00	0 85	0 74	0 64
Octane (n)	43 6	1 18	1 09	1 00	0 91	0 85	0 78
Pentane (n)	39 2	1 27	1 12	1 00	0 91	0 85	0 80
Propyl alcohol (iso)	48 9	1 61	1 25	1 00	0 84	0 73	0 64
Sulfur dioxide	86 0	1 21	1 10	1 00	0 90	0 81	0 73
Sulfuric acid (60%)	93 5		1 18	1 00	0 89	0 97	0 63
Toluene	53 9	1 20	1 16	1 00	0 92	0 85	0 79
Water	62 3		1 17	1 00	0 89	0 81	0 74

* Average temperature of the film = (temperature of the tube wall + temperature of the main body of the liquid) ÷ 2

† Values in italics are based on extrapolated values of the physical properties of the fluids.

TABLE 48—DIAMETER-CORRECTION FACTOR FOR CASE 1

Inside tube dia in.	F_d	Inside tube dia in.	F_d
0 2	7 20	1 2	0 80
0 3	4 37	1 3	0 72
0 4	3 08	1 4	0 66
0 5	2 34	1 5	0 61
0 6	1 88		
0 7	1 55	2 0	0 43
0 8	1 31	2 5	0 32
0 9	1 14	3 0	0 25
1 0	1 00	3 5	0 22
1 1	0 89	4 0	0 18

TABLE 49—MINIMUM PERMISSIBLE VALUE OF $V \times d$ FOR CASE 1

V = velocity, ft/sec d = inside tube dia, in.

Liquid*	Ave temp of liquid $(V \times d)_{min}$				
	50°F	100°F	150°F	200°F	250°F
Acetic acid (50%)	1 4	0 7	0 4	0 3	0 2
Amyl alcohol (iso)	3 8	1 6	0 8	0 5	0 3
Aniline	3 0	1 5	0 8	0 4	0 3
Butyl alcohol (n)	2 4	1 2	0 7	0 4	0 3
Ethyl alcohol (100%)	1 0	0 6	0 4	0 3	0 2
Ethyl alcohol (40%)	2 4	0 9	0 5	0 4	0 3
Ethylene glycol (50%)	4 5	1 0	0 5	0 3	0 2
Glycerol (50%)	3 0	1 7	0 0	0 5	0 4
Methyl alcohol (40%)	1 4	0 5	0 3	0 2	0 2
Propyl alcohol (iso)	2 4	0 0	0 5	0 4	0 3
Sulfuric acid (60%)	8 6	1 4	0 9	0 7	0 7

* Only those liquids for which $(V \times d)_{min}$ at 100°F is greater than 0 5 are listed.

Liquid	Curve No.
Acetic acid (100%)	5
Acetic acid (50%)*	4
Acetone	10
Ammonia	14
Amyl acetate	8
Amyl alcohol (Iso)*	5
Aniline*	3
Benzene	9
Brine (CoCl₂)(25%)	4
Butyl alcohol (N)*	6
Carbon disulfide	6
Carbon tetrachloride	3
Chlorobenzene	6
Chloroform	4
Ethyl acetate	9
Ethyl alcohol (100%)*	8
Ethyl alcohol (40%)*	4

Liquid	Curve No.
Ethyl bromide	5
Ethylene glycol (50%)*	3
Ethyl ether	12
Ethyl iodide	2
Glycerol (50%)*	2
Heptane	11
Hexone	12
Methyl alcohol (100%)	9
Methyl alcohol (90%)	9
Methyl alcohol (40%)*	6
Octane (N)	10
Pentane (N)	13
Propyl alcohol (Iso)*	7
Sulfur dioxide	6
Sulfuric acid (60%)*	1
Toluene	9
Water	6

*Chart not valid if V×d is less than minimum value given in Table 49

$\left(\dfrac{\Delta p}{N}\right)_0$ = Base value of the pressure drop per foot of length, lb. per sq.in. per foot of length

V = Velocity, ft. per sec

Fig. 39.—Base value of the pressure drop for Case 1.

Case 2. Petroleum Oils, Pressure Drop inside Tubes or Pipes, Streamline Flow —The pressure drop per foot of length can be determined by the equation

$$\left(\frac{\Delta p}{\lambda}\right) = \left(\frac{\Delta p}{\lambda}\right)_o \times F_d$$

where $\left(\frac{\Delta p}{\lambda}\right)$ = the pressure drop per foot of length, lb/(sq in.)(ft of length)

$\left(\frac{\Delta p}{\lambda}\right)_o$ = the base value of the pressure drop per foot of length, from Fig. 40

F_d = the diameter-correction factor, from Table 50

TABLE 50—DIAMETER-CORRECTION FACTOR FOR CASE 2

Inside Tube Diam., in.	F_d*
0.5	16.00
0.6	7.72
0.7	4.19
0.8	2.44
0.9	1.52
1.0	1.00
1.1	0.64
1.2	0.48
1.3	0.35
1.4	0.26
1.5	0.20

* If greater accuracy is required than can be obtained by interpolation, F_d may be calculated by the equation

$$F_d = \frac{1}{d^4}$$

where d = the inside tube diameter in.

TABLE 51—MAXIMUM PERMISSIBLE VALUES OF W/d FOR CASE 2

W = rate of flow per tube lb per hr per tube d = inside tube diameter in.

Viscosity Saybolt Universal seconds	GRAVITY °A.P.I.					
	10°	20°	30°	40°	50°	60°
	$\left(\frac{W}{d}\right)_{max}$					
40	1,300	1,400	1,500	1,600	1,700	1,800
50	2,300	2,500	2,700	2,800	3,000	3,200
60	3,200	3,500	3,700	3,900	4,100	4,400
80	4,800	5,200	5,500	5,800	6,200	6,500
100	6,400	6,800	7,200	7,800	8,200	8,700
150	10,100	10,900	11,600	12,300	13,000	13,700
200	13,800	14,800	15,700	16,700	17,700	18,700
300	20,600	22,100	23,400	25,000	26,400	27,900
400	27,600	29,600	31,500	33,400	35,400	37,400

* The viscosity should be calculated at an average temperature equal to $(3t + t_w)/4$, where t is the temperature of the oil and t_w is the temperature of the tube wall

FIG. 40.—Base value of the pressure drop for Case 2

Case 3   Liquids, Pressure Drop at the Entrance to Tubes.—The pressure drop at the entrance to unflared tubes (or at any other abrupt contraction in cross section) can be determined by the equation

$$\Delta p = \Delta p_b \times F_r,$$

where Δp = the pressure drop, lb per sq in

Δp_b = the base value of the pressure drop, from Fig 41

F_r = the area-ratio-correction factor, from Table 52
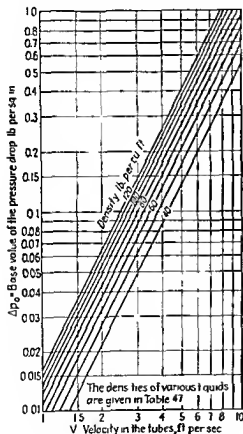
TABLE 52—AREA-RATIO-CORRECTION FACTOR
FOR CASE 3

A_1 = cross-sectional area before entering the tubes

A_2 = cross-sectional area of the tubes.

$\dfrac{A_1}{A_2}$	F_r
1 0	0 00
1 5	0 54
2 0	0 75
2 5	0 86
3 0	0 92
3 5	0 97
4 0	1 00
6 0	1 07
8 0	1 11
10 0	1 13

Fig 41—Base value of the pressure drop for Case 3

Case 4. Liquids, Pressure Rise at the Outlet from Tubes.—The *rise* in pressure at the outlet from tubes (or at any other abrupt enlargement in cross section) can be determined by the equation

$$\Delta p = \Delta p_0 \times F_r,$$

where Δp = the *rise* in pressure, lb. per sq. in.

Δp_0 = the base value of the rise in pressure, from Fig. 42.

F_r = the area-ratio-correction factor, from Table 53.
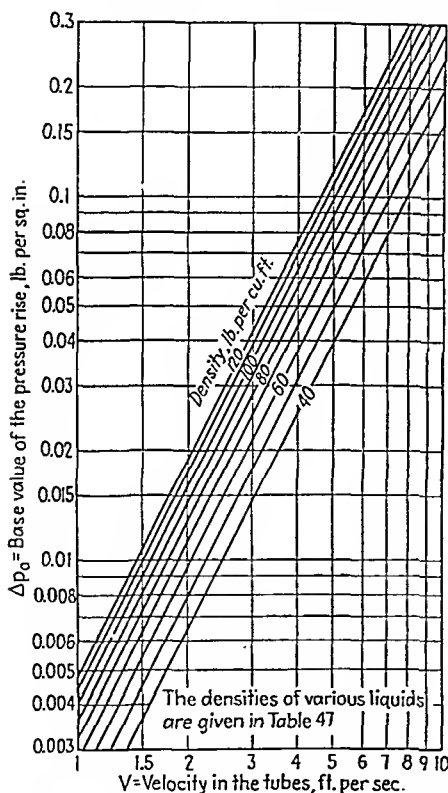
TABLE 53.—AREA-RATIO-
CORRECTION FACTOR
FOR CASE 4

A_1 = cross-sectional area
of the tubes.

A_2 = cross-sectional area
after leaving the
tubes.

$\dfrac{A_2}{A_1}$	F_r
1.0	0.00
1.5	1.18
2.0	1.33
2.5	1.28
3.0	1.18
3.5	1.09
4.0	1.00
6.0	0.74
8.0	0.58
10.0	0.48

FIG. 42.—Base value of the pressure rise for Case 4.

Case 5 Gases, Pressure Drop inside Smooth Tubes, Turbulent Flow —
The pressure drop per foot of length can be determined by the equation

$$\left(\frac{\Delta p}{N}\right) = \left(\frac{\Delta p}{N}\right)_s \times F_t \times F_d \times \frac{1}{P}$$

where $\left(\frac{\Delta p}{N}\right)$ = the pressure drop per foot of length, lb /(sq in)(ft of
length)

$\left(\frac{\Delta p}{N}\right)_s$ = the base value of the pressure drop per foot of length, from
Fig 43

F_t = the temperature-correction factor, from Table 54
F_d = the diameter-correction factor, from Table 55
P = the average pressure of the gas atm abs (1 atm = 14 7 lb
per sq in)

TABLE 54 —TEMPERATURE-CORRECTION FACTOR FOR CASE 5		TABLE 55 —DIAMETER CORRECTION FACTOR FOR CASE 5	
Ave Temp of Gas °F	F_t	Inside Tube Dia In	F_d
100	0 60	0 2	7 20
0	0 80	0 3	4 37
100	1 00	0 4	3 08
200	1 21	0 5	2 34
300	1 42	0 6	1 88
400	1 64	0 7	1 55
500	1 86	0 8	1 31
		0 9	1 14
		1 0	1 00
		1 1	0 89
		1 2	0 80
		1 3	0 72
		1 4	0 66
		1 5	0 61
		2 0	0 43
		2 5	0 32
		3 0	0 25
		3 5	0 22
		4 0	0 18

Gas	Curve No
Acetylene	5
Air	5
Ammonia	4
Butane	10
Carbon dioxide	8
Carbon monoxide	5
Chlorine	11
Ethane	6
Ethylene	6

Helium	2
Hydrogen	1
Hydrogen sulfide	7
Methane	3
Methyl chloride	9
Nitric oxide	5
Nitrogen	5
Nitrous oxide	8
Oxygen	6
Sulfur dioxide	10

$\left(\dfrac{\Delta p}{N}\right)_0$ = Base value of the pressure drop per foot of length, lb per sq in per ft of length

$G = \dfrac{\text{Rate of flow}}{\text{Cross sectional area}}, \dfrac{\text{lb per sec}}{\text{sq ft}} = V\rho = \text{velocity} \times \text{density, (ft per sec.)} \times \text{(lb per cu ft)}$

FIG 43 —Base value of the pressure drop for Case 5.

Case 6. Gases, Pressure Drop at the Entrance to Tubes.—The pressure drop at the entrance to unflared tubes (or at any other abrupt contraction in cross section) can be calculated by the equation

$$\Delta p = \Delta p_o \times F_r,$$

where Δp = the pressure drop lb per sq in

Δp_o = the base value of the pressure drop from Fig 44

F_r = the area ratio-correction factor from Table 56

TABLE 56 — AREA RATIO-CORRECTION FACTOR FOR CASE 6

A_1 = cross-sectional area before entering the tubes

A_2 = cross-sectional area of the tubes

$\dfrac{A_1}{A_2}$	F_r
1 0	0 00
1 5	0 54
2 0	0 75
2 5	0 86
3 0	0 92
3 5	0 97
4 0	1 00
6 0	1 07
8 0	1 11
10 0	1 13

TABLE 57 — GAS DENSITIES

Approximate values of the density can be calculated by the equation

$$\rho = \frac{bp}{t + 460},$$

where ρ = the density of the gas lb per cu ft

p = the absolute pressure of the gas lb /sq in abs

t = the temperature of the gas °F

b = a constant obtained from the following table

Gas	b
Acetylene	2 46
Air	2 70
Ammonia	1 61
Butane	5 57
Carbon dioxide	4 13
Carbon monoxide	2 61
Chlorine	6 72
Ethane	2 83
Ethylene	2 63
Helium	0 37
Hydrogen	0 19
Hydrogen sulfide	3 21
Methane	1 50
Methyl chloride	4 82
Nitric oxide	2 80
Nitrogen	2 61
Nitrous oxide	4 13
Oxygen	2 98
Sulfur dioxide	6 11
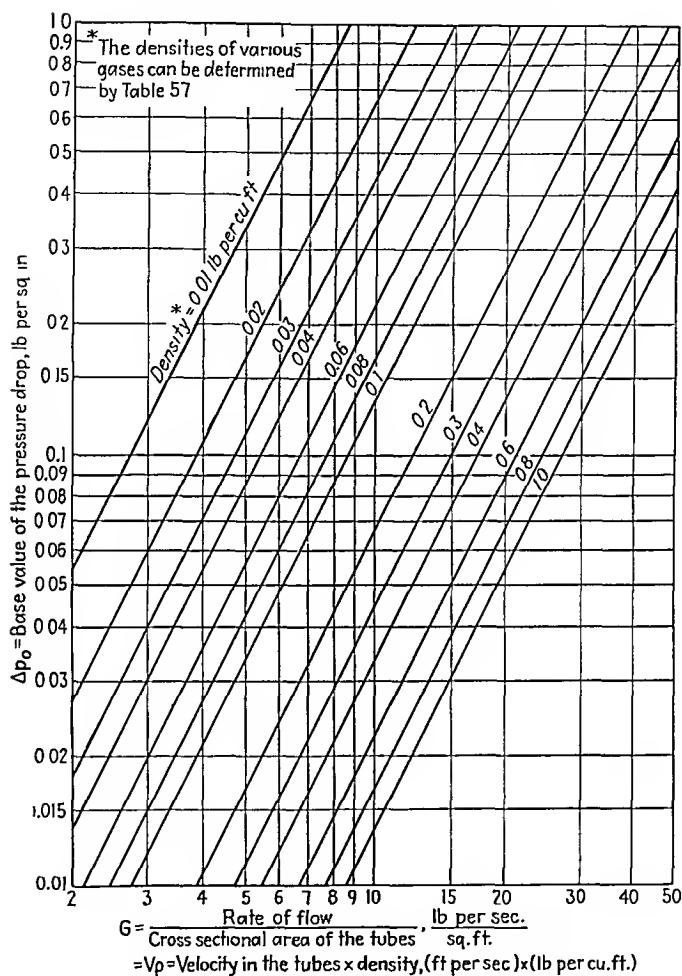
FIG. 44.—Base value of the pressure drop for Case 6.

Case 7. Gases, Pressure Rise at the Outlet from Tubes.—The *rise* in pressure at the outlet from tubes (or at any other abrupt enlargement in cross section) can be determined by the equation

$$\Delta p = \Delta p_b \times F_r$$

where Δp = the *rise* in pressure, lb per sq in

Δp_b = the base value of the rise in pressure, from Fig 45

F_r = the area ratio-correction factor, from Table 58

TABLE 58.—AREA RATIO-CORRECTION FACTOR FOR CASE 7

A_1 = cross-sectional area of the tubes

A_2 = cross sectional area after leaving the tubes

$\dfrac{A_2}{A_1}$	F_r
1 0	0 00
1 5	1 18
2 0	1 33
2 5	1 28
3 0	1 18
3 5	1 09
4 0	1 00
6 0	0 74
8 0	0 58
10 0	0 48

Fig. 45 —Base value of the pressure drop for Case 7
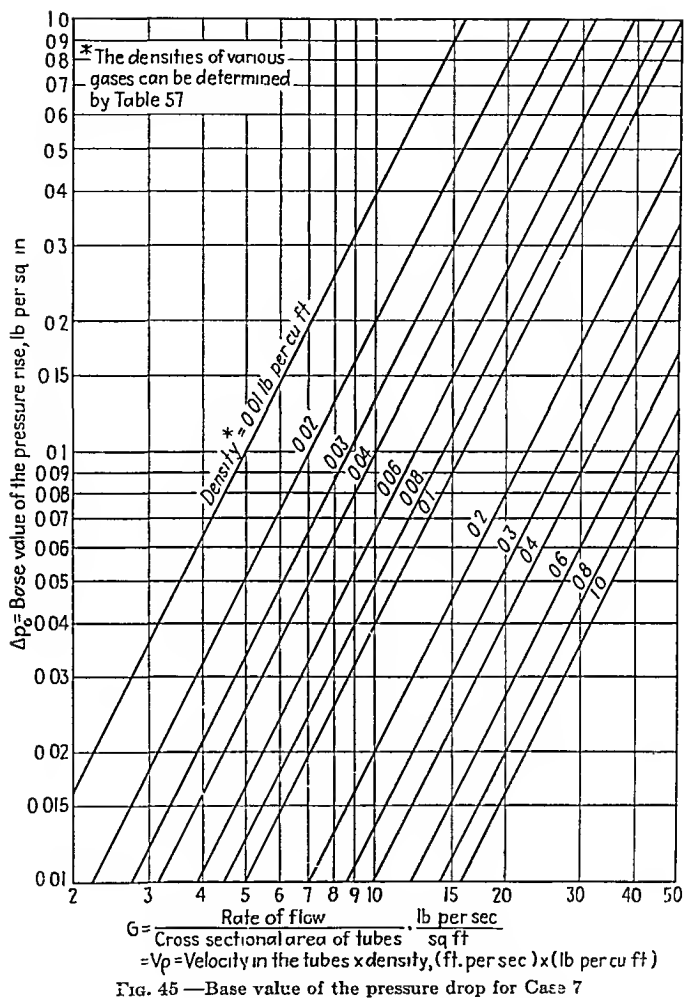
Case 8   Gases, Pressure Rise or Drop Caused by Changes in Density —
The change in pressure caused by changes in the density of a gas flowing
inside tubes can be determined by the equation

$$\Delta p = p_0{}' - p_0{}',$$

where Δp = the drop in pressure if positive and the rise in pressure if negative, lb per sq in

$p_0{}'$ = the pressure term for the gas entering the tubes, from Fig 46

$p_0{}''$ = the pressure term for the gas leaving the tubes, from Fig 46

Note that Δp is positive if the gas is heated and negative if cooled

Case 9   Pressure Drop Caused by a Reversal in the Direction of Flow —
The pressure drop caused by a reversal in the direction of flow depends upon
the design of the passage in which the reversal takes place   If the passage
is well rounded the pressure drop is usually negligible, but if the passage
has sharp corners, as in the headers of a multipass heat exchanger, the
pressure drop may be relatively large

For design purposes the energy loss in passages with sharp corners may
be assumed approximately equal to one-half the kinetic energy of the fluid
in the tubes   Based on this assumption, the pressure drop can be estimated
by this equation

$$\Delta p = 0.0000031\, V^2\rho, \qquad (58)$$

where Δp = the pressure drop   lb per sq in

V = the velocity in the tubes, ft per sec

ρ = the density of the fluid   lb per cu ft

**Case 10   Liquids or Gases, Pressure Drop in Annular Spaces or outside
Tube Bundles, Direction of Flow Parallel to Tubes** —The pressure drop per
foot of length can be determined from Case 1 or 5 for fluids *inside* tubes if
an equivalent diameter d_s is used in determining the diameter correction
factor F_d The correct equivalent diameter for flow outside tube bundles
can be determined either by Eq (46) or by Fig 29 or 30 given in Case 9 in
the preceding chapter   The correct equivalent diameter for annular spaces
is equal to the difference between the outside and the inside diameters of the
annular space
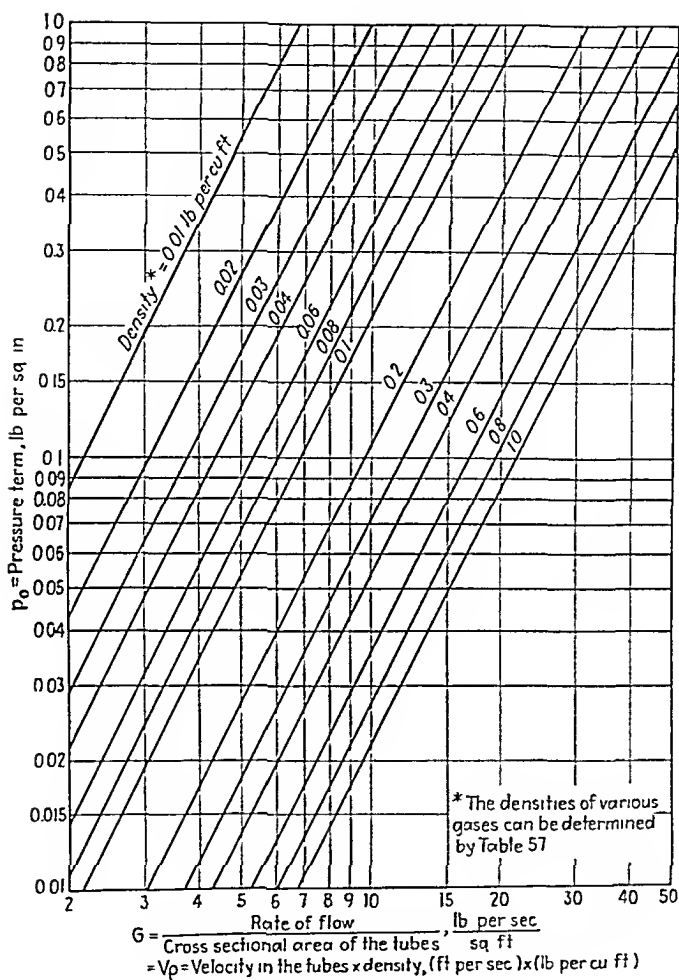
FIG 16 —Pressure term for Case S

CHAPTER VI

TYPES OF INDUSTRIAL HEAT-TRANSFER EQUIPMENT

36 Introduction—There are many types of heat-transfer equipment Some of the more common types used for transfer ring heat from one fluid to another are described in this chapter Direct-fired or electrically heated equipment is not included

37 Shell-and-tube-type Equipment—*a* One of the most widely used types of heat transfer equipment consists of a num-
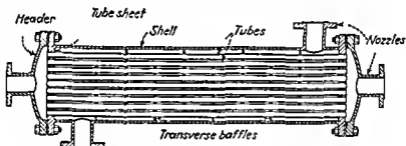
Fig 47 —Cross section of a simple shell and tube heat exchanger

ber of parallel tubes inclosed in a relatively close-fitting cylin drical shell One fluid flows *inside* the tubes and is called the *tube side* fluid the other flows outside the tubes and is called the *shell side* fluid If neither fluid condenses or evaporates the equipment is commonly designated as a *heat exchanger* If one of the fluids condenses the equipment is designated either as a *condenser* or as a *heater* depending upon whether the primary purpose of the equipment is to condense the one fluid or to heat the other Similarly if one of the fluids evaporates the equip ment is designated either as an *evaporator* or as a *cooler* depending upon whether the primary purpose is to evaporate the one fluid or to cool the other

One of the simplest designs is shown in Fig 47 The tube sheets are an integral part of the shell and both fluids make a single pass through the equipment The transverse baffles shown are not essential but they increase the velocity and

turbulence of the shell-side fluid and thereby increase the rate of heat transfer. They are omitted if the shell-side fluid condenses or evaporates or if the volume of flow is large. The size of the nozzles used depends upon the service for which the equipment is intended. For example, the vapor-inlet nozzles on condensers intended for vacuum service are usually made quite large, are not necessarily circular in cross section, and may extend almost the entire length of the shell.

Although the design shown in Fig. 47 is simple, it has certain disadvantages. Thus, no provision is made to allow for differences in the expansion of the tubes and of the shell. Consequently, the design is not suitable if the temperature difference
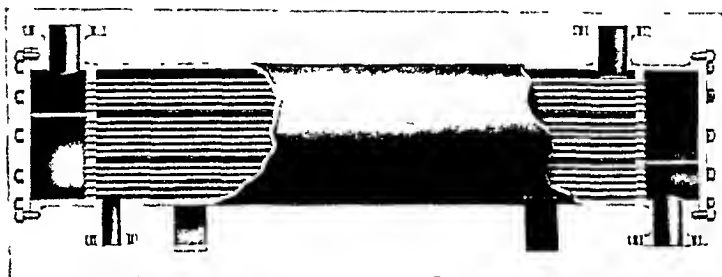
Fɪɢ. 48.—Cutaway view of a shell-and-tube heat exchanger arranged for multipass flow of the tube-side fluid. (*Courtesy of the Griscom-Russell Co.*)

between the tube-side and shell-side fluids is large. Moreover, since the outside surfaces of the tubes are not accessible for cleaning, the design cannot be used if the shell-side fluid is dirty or scale forming. Finally, the headers cannot be removed in order to clean the inside surfaces of the tubes without breaking the piping connections.

Another relatively simple design is shown in Fig. 48. This is similar to the one shown in Fig. 47 in that (1) no provision is made to allow for differences in the expansion of the tubes and of the shell and (2) the tube sheets are an integral part of the shell; so the outside surfaces of the tubes are not accessible for cleaning. This design offers a distinct improvement over the preceding one, however, in that the inside surfaces of the tubes can be readily cleaned without disturbing any of the piping connections. Note that the headers are provided with partitions

which cause the tube-side fluid to make three passes through
the exchanger

Many variations of these two simple designs are possible (1)
by using various kinds of tubes, (2) by using various kinds of
baffles in the shell, (3) by providing partitions in the covers to
produce multipass flow, and (4) by including some provision to
allow for differences in the expansion of the tubes and of the
shell

b   Tubes—Straight, bare tubes are most commonly used, but
for certain applications other kinds are advantageous Straight
tubes set into the tube sheets with a slight initial bow or curva-
ture are useful when the shell-side fluid is scale forming With
such tubes changes in temperature cause variations in flexure

Fig 49—Cross section of a shell and tube heat exchanger with U tubes

which tend to crack up and break off scale accumulations
Bowed tubes also provide for expansion and contraction but
they should not be used in short lengths because of the increased
likelihood of tube failure at the tube sheets

U, or "hairpin" tubes illustrated in Fig 49 are used to take
care of expansion and contraction These require only one tube
sheet, but single defective tubes are sometimes accessible for
replacement only by removing several other tubes Since the
tubes are difficult to clean internally they are suitable for use
only when the tube-side fluid is clean and nonfouling

Finned tubes are particularly adapted for use when the shell
side fluid has a much lower film coefficient than the tube-side
fluid, *e g*, when a gas is to be heated or cooled by a liquid Either
transverse or longitudinal fins, as shown in Fig 50 can be used
depending upon whether the flow is normal to or parallel to the
tubes

Retarder, or "swirling," strips, consisting of a twisted metal strip inserted inside the tubes, are rarely used. Better results can be obtained with the same pressure drop by using either

(a)

(b)
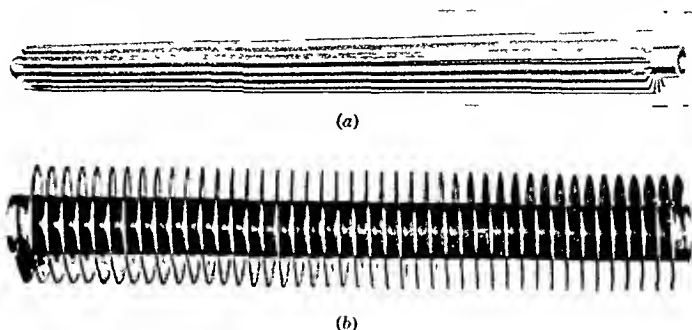
FIG. 50.—Finned tubes. (a) Longitudinal fins. (b) Transverse fins. (*Courtesy of the Griscom-Russell Co.*)

longer or smaller diameter tubes or by increasing the number of passes.

c. *Baffles.*—There are two general types of baffles: transverse and longitudinal. Both are used to increase the rate of heat transfer by increasing the velocity and turbulence of the shell-side fluid. They add considerably to the cost of the equipment,
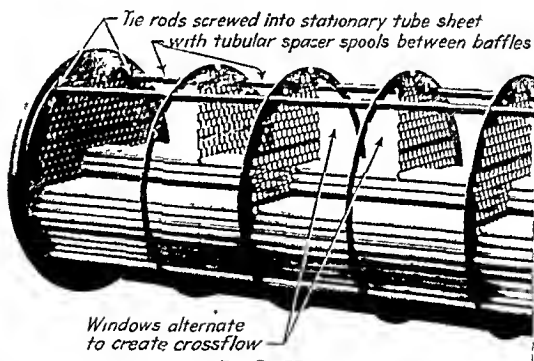
FIG. 51.—Partially assembled tube bundle showing method of holding transverse baffles in place. (*Courtesy of the Ross Heater and Mfg. Co.*)

however, and to the difficulty of cleaning the outside surface of the tubes. They also increase the pressure drop through the shell. In order to be fully effective, they must be installed so

that by-passing is eliminated as far as possible This is accomplished by reducing to a minimum the clearance between the baffles and the shell and between the baffles and the tubes Sufficient clearance must be allowed, however, to permit the tube bundle to be withdrawn readily, even after some warping has taken place The baffles are held in place independent of the tubes by tie rods and spacers, as shown in Fig 51 It is important that the baffles be supported rigidly in order to prevent vibration which would result in chafing and quick failure of the tubes

(a) Segmental type

(b) Disc and ring type

(c) Orifice type

(d) Vane type
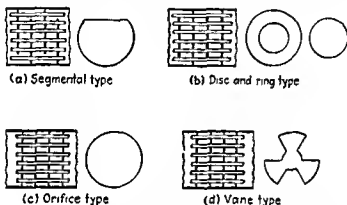
Fig 52 —Types of transverse baffles (tube holes not shown)

Several kinds of transverse baffles are used, probably the commonest being the *segmental* or "cross flow," type shown in Fig 52a This type of baffle causes the shell-side fluid to flow back and forth across the tube bundle and is suitable for use with fouling liquids or for large volumes of flow

Disk-and-ring baffles shown in Fig 52b, force the fluid to flow successively from the outside of the tube bundle toward the center and back again to the outside These baffles are slightly more expensive than the segmental type and have no particular advantage over them

Orifice baffles, shown in Fig 52c, force the fluid to flow at high velocities through the annular space between the baffles and the tubes They can be used for small or moderate volumes of flow

Vane-type baffles, shown in Fig 52d, are slightly twisted like the blades of a fan By turning the baffles so that the blades of

one are in line with the openings of the next, a moderate deflection of the fluid is obtained. These baffles provide a large area for flow and consequently cause only a small pressure drop. They are particularly suitable for use with gases.

Helical baffles are made of a number of disks which are cut along one radius and welded together to form a continuous helix. They are more expensive than any of the other types and have no outstanding advantages over them.

Longitudinal baffles consist of flat plates which extend all the way across the shell. They may be used to obtain counter-flow of the tube-side and shell-side fluids, as shown in Fig. 53. They are also frequently used in condensers to distribute the flow of vapor over the entire length of the tube bundle when
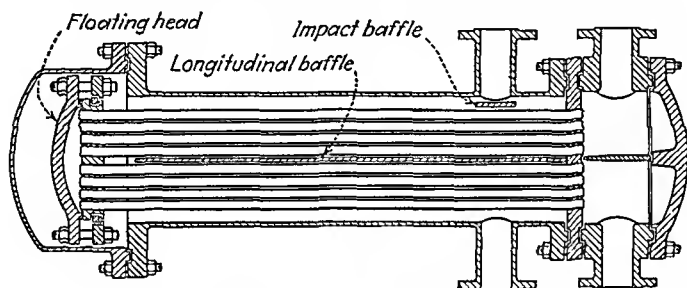
Fig. 53.—Cross section of a shell-and-tube heat exchanger equipped with a full floating head.

the vapor-inlet nozzle is located in the center of the shell. The clearance between the baffle and the shell should be as small as possible in order to prevent by-passing, but at the same time it must be large enough to permit the tube bundle to be withdrawn easily. If a split floating head is used, the longitudinal baffle may be permanently welded to the shell.

Impact baffles, as shown in Fig. 53, are placed opposite the shell-inlet nozzle. They are used to prevent erosion, caused by impingement of the incoming fluid on the tubes, and to prevent tube vibration. They also help to distribute the flow.

d. Passes.—If the tube-side fluid passes from one end of the heat exchanger to the other end just once, as in Fig. 47, the exchanger is said to be *single-pass* on the tube side. In contrast, the heat exchangers shown in Figs. 48 and 53 are said to be three-pass and two-pass, respectively. By adding suitable partitions

inside the headers, the tube-side fluid can be made to make as many passes as may be desired Such *multipass* arrangements are used to obtain higher velocities and longer paths for the fluid to travel without increasing the length of the exchanger

e Expansion Provisions—It is frequently necessary to make some provision to allow for differences in the expansion of the tubes and of the shell It is sometimes also necessary to provide for differences in the expansion of the various tubes if either the tube-side or the shell side fluid undergoes a large change in
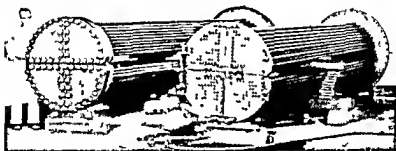
Fig 54—Tube bundles with split floating heads   *(Courtesy of the Henry Vogt Machine Co)*

temperature The following methods are used to provide for these differences in expansion

The *full floating-head* construction, shown in Fig 53, is relatively expensive but is particularly suited for use when the shell-side fluid is under high pressure The tube bundle can be removed readily for cleaning the outside surface of the tubes or for making repairs, and since straight tubes are used the inside surface of the tubes can also be easily cleaned The channel type of cover shown on the stationary head permits easy access to the inside of the tubes for cleaning or inspection without requiring the piping connections to be broken

The *split floating-head* construction shown in Fig 54 is used if either of the fluids passing through the exchanger undergoes a large change in temperature The tube-side fluid is made to pass successively through each of the floating heads, and the differences in the temperatures of the tubes connected to any one floating head are thereby reduced

The *packed floating-head* construction, shown in Fig 55, also permits free expansion of the tubes and the inside surface

of the tubes are readily accessible for cleaning by removing the cover plates on the stationary and floating heads. The entire tube bundle can be withdrawn to allow access to the outside surfaces of the tubes. The exchanger shown is two-pass on the tube side; so one row of tubes is omitted in the center of the tube
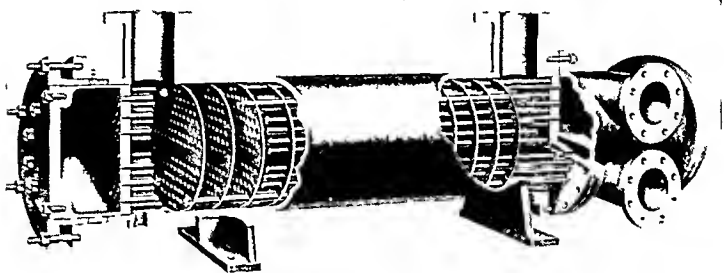
Fig. 55.—Cutaway view of a shell-and-tube heat exchanger equipped with orifice-type baffles and a packed floating head. (*Courtesy of the Griscom-Russell Co.*)

bundle because of the partition required in the stationary head. Note the tie rods and spacer spools used to hold the orifice-type baffles in place.

A modification of the packed floating-head construction is shown in Fig. 56. A smaller packing gland is required, and single-pass flow through the tubes is possible. This design is rarely used, however, because it is as expensive as either the full floating head or packed floating head without having any outstanding advantages over them. Note that the expansion of the tubes is transmitted to the external piping and that the latter must be arranged to take care of this movement.
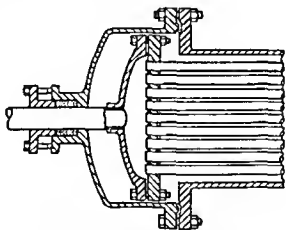
Fig. 56.—Cross section showing packing gland on pipe connected to floating head.

Differences in the expansion of the tubes and of the shell are frequently provided for by some sort of flexible section built into the shell. Three such designs are shown in Fig. 57. These designs are suitable for moderate differences in expansion.

The *U tubes* shown in Fig. 49 offer a relatively inexpensive means of providing for expansion. Since the tubes are free to

expand independently of each other, they are particularly suited for use where the various tubes may be subjected to widely differing temperatures They can be used for any pressure, but because of the difficulty of cleaning them internally, they should not be used when the tube-side fluid is dirty or scale forming *Bowed tubes* can also be used to take care of expansion, but

(a)   (b)   (c)
Corrugated or rolled into shell   Flexible flanges in shell   Flexible flange of tube sheet

Fig 57—Types of flexible sections built into the shell

because of the repeated flexing they are likely to fail or develop leaks at the tube sheet unless they are relatively long

Individually packed tubes each tube provided with a stuffing box and *ferrule* as shown in Fig 58 are frequently used in steam power plant condensers This arrangement allows each tube to expand independently but involves a large number of joints to be kept tight
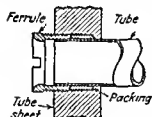
Fig 58—Individually packed tube

Another arrangement that permits independent tube expansion consists of two concentric tubes One end of the outer tube is plugged and the two tubes are expanded into separate tube sheets The tube-side fluid flows through the smaller tube to the plugged end of the larger and back through the annular space between them Heat transfer between the shell-side fluid outside the larger tube and the tube-side fluid inside takes place while the latter is in the annular space

38 Double-pipe Heat Exchangers—Double-pipe heat exchangers are sometimes used in place of shell and tube heat exchangers for low rates of flow They usually consist of two concentric pipes one of the fluids flowing through the inner pipe and the other flowing in the opposite direction through the

annular space between the pipes.   True counterflow is obtained, but this advantage is somewhat offset by the relatively large outside surface requiring insulation.   On the basis of cost per square foot of heat-transfer surface, these exchangers are relatively expensive and consequently are usually used only for special applications.
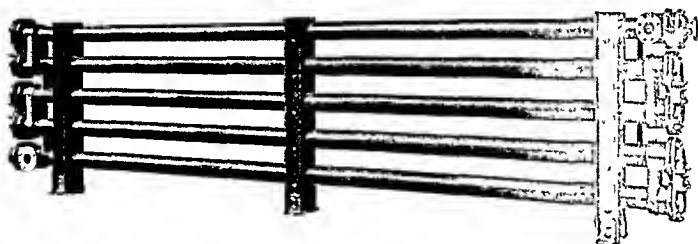
Fig. 59.—Double-pipe heat exchanger.   *(Courtesy of the Struthers-Wells Co.)*

A typical double-pipe heat exchanger is shown in Fig. 59. Expansion is provided for by the packing gland on the inner pipe.

A special design of double-pipe heat exchanger, intended for use when a gas is to be heated or cooled by a liquid, is shown in Fig. 60.   The longitudinal fins on the inner pipe compensate

Fig. 60.—Double-pipe heat exchanger equipped with longitudinal fins.   *(Courtesy of the Griscom-Russell Co.)*

for the low film coefficient of the gas and reduce the length of pipe required.

A rather unusual design is shown in Fig. 61.   Instead of the tubes being concentric, they are expanded throughout their entire length into fins, heat flowing from one fluid to the other through the fins.   The inlet and outlet connections and the return bends

type has 2-in. diameter tubes about 5 ft. long, although tubes from 1 to 4 in in diameter and from 3 to 6 ft. long are common.

c. Basket-type Evaporator.—The basket-type evaporator, shown in Fig 64, operates in the same manner as the standard vertical short-tube evaporator except that the downtake is annular The deflector helps reduce entrainment losses and can be easily installed. One of the advantages of the basket-type
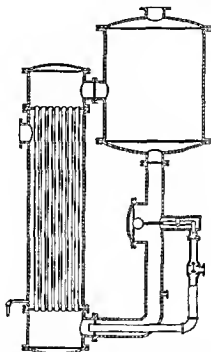
Fig 65 —Vertical long tube evaporator

evaporator is the fact that the entire heating element can be removed for cleaning and repairing About the same diameter and length tubes are used as in the standard short-tube evaporator.

d. Vertical Long-tube Evaporator —Vertical long-tube evaporators are similar to vertical short-tube evaporators, except that the tubes are usually from 10 to 20 ft long These evaporators can be divided into two types the controlled-level type, shown in Fig 65, and the film type In the controlled-level type the liquor level is maintained at from one-third to one-half the height

of the tubes by a float-controlled feed valve in the downtake. The vapor discharges either into a vapor head directly above and integral with the heating element or into a separate vapor head as shown. Baffles are used to help separate the entrained liquor from the vapor.

In the film-type evaporator the liquor level is not controlled, and the downtake pipe is used only in starting. After the
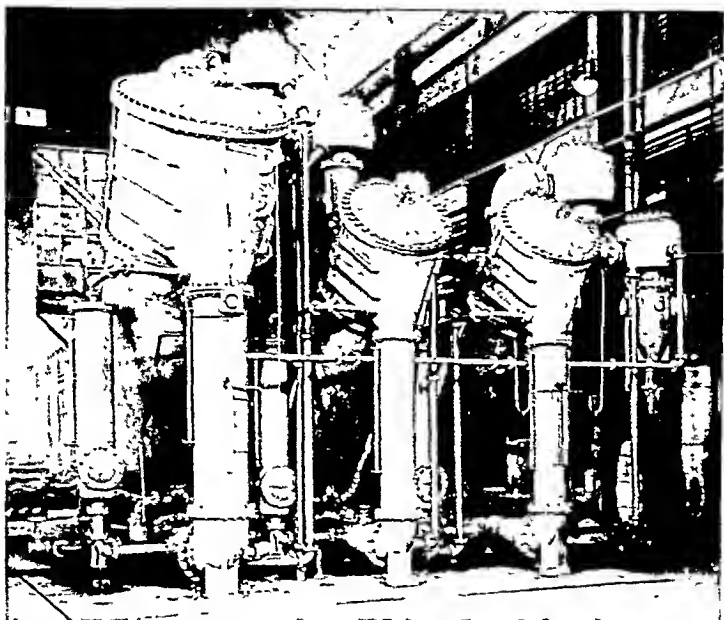
Fig. 66.—Triple-effect inclined-tube evaporator. (*Courtesy of the Struthers-Wells Co.*)

evaporator is in operation, the liquor makes only a single pass through the tubes, the vapor and entrained liquid discharging usually into a centrifugal-type separator. The liquid carried over into the separator is withdrawn through a float-controlled valve which is required to prevent by-passing of the vapor.

  e. Inclined-tube Evaporator.—Inclined-tube evaporators are similar to vertical long-tube evaporators except that the tubes are inclined, usually at an angle of 45 deg. This type of evaporator requires less headroom than the vertical type, and tube replacements are more easily made. Since these evaporators can

be made self-supporting, they require no structural framework and are relatively easy to install. A large triple-effect evaporator of this type is shown in Fig. 66. Note the downtake pipes and liquid level controllers.

f. Horizontal-tube Evaporators.—Horizontal-tube evaporators differ from vertical- and inclined-tube evaporators in that the
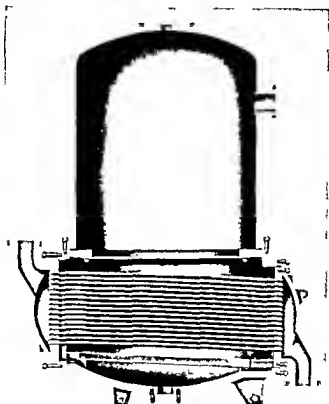
Fig. 67.—Sectional view of a vertical-body horizontal-tube evaporator. (*Courtesy of the Griscom-Russell Co.*)

liquor is outside the tubes and the steam inside. They are made with either rectangular or cylindrical bodies. The rectangular type of body is often built of flat cast-iron plates and is known as the *Wellner-Jelinek type.* This construction permits easy access for repairs but involves a large number of joints to be kept tight. The cylindrical type of body is built with the body either vertical or horizontal. Figure 67 shows a vertical-body

evaporator equipped with bowed tubes which, when subjected
to changes in temperature, tend to crack off and shed scale
automatically. The horizontal-body evaporator shown in Fig.
68 is also equipped with bowed tubes and is designed for pro-
ducing boiler-feed make-up water. The entire tube bundle can
be withdrawn, and the inside surfaces of the tubes can be cleaned
by unscrewing the plugs in the headers opposite the ends of each
tube.

g. Forced-circulation Evaporator.—The forced circulation evap-
orator is similar to the long-tube vertical evaporator, except
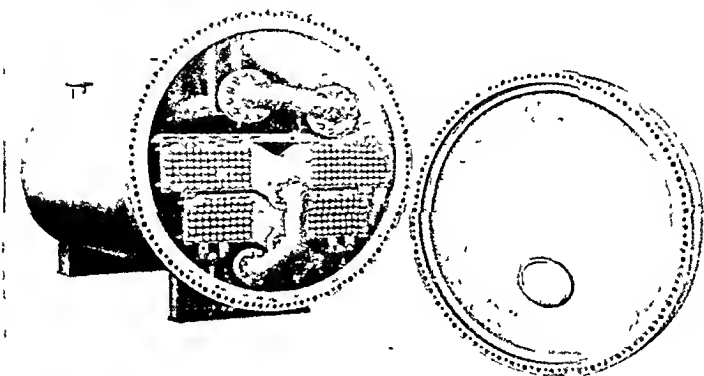
Fig. 68.—Horizontal-body, horizontal-tube evaporator with cover removed.
(Courtesy of the Griscom-Russell Co.)

that a centrifugal pump is placed in the circuit between the
downtake and the tube bundle, as shown in Fig. 69. This
permits higher velocities, and consequently higher rates of heat
transfer, to be obtained. Since less heating surface is required,
this type of evaporator is particularly adapted to applications
that require that the heating surface be made of some high-priced
metal. The liquor, which is inside the tubes, emerges from
the heating element at high velocities and is directed downward
by a baffle into the vapor head. The feed and thick-liquor take-
off connections can be made at any point in the circuit, although
admitting the feed on the inlet side of the pump is advantageous
in that the feed and thick liquor are thoroughly mixed on passing
through the pump.

h Coiled-tube Evaporator—Another type of evaporator intended for producing boiler feed make up water is shown in Fig 70 The heating surface consists of helical coiled tubes which, like bowed tubes, crack up and break off scale accumulations automatically when subjected to variations in temperature

41. Open-type Sections—Open-type sections are characterized by the fact that the tube bundle is not inclosed in any kind of shell A typical open-type section is shown in Fig 71 These sections are made with either bare or finned tubes and can be arranged for multipass flow of the tube-side fluid by providing partitions in the headers

Bare-tube sections are frequently used as condensers or as gas or liquid coolers For either of these applications, the sections can be submerged in open tanks or pits through which the cooling water flows Baffles may be provided in the tanks to obtain counterflow of the cooling water If a limited water supply makes atmospheric cooling necessary, the sections can be suspended directly beneath cooling towers, and the cooling water allowed to drip over them Bowed tubes are used if the
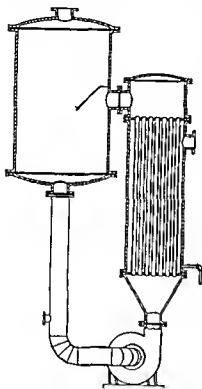
Fig 69 —Forced-circulation evaporator

sections are subjected to large variations in temperature or if the cooling water is scale forming

Bare-tube sections are often used for recovering heat from hot waste water in plants, such as dye houses, woolen mills, bleacheries, and laundries The sections are placed in pits through which the hot waste water flows while the clean water to be heated flows through the tubes This arrangement is particularly advantageous if the waste water contains lint or other

solid matter. Such impurities would quickly clog the passages of a shell-and-tube heat exchanger, but they can easily be washed off open sections with a hose.

Bare-tube sections are sometimes also used for heating chemicals or other solutions. The sections are submerged in the solution, and the steam for heating flows through the tubes.
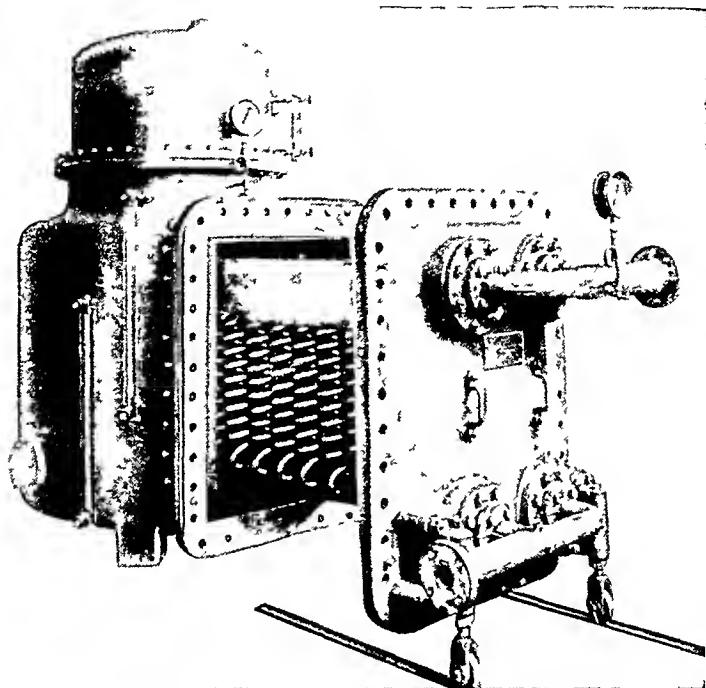
Fig 70—Coiled-tube evaporator   *(Courtesy of the Griscom-Russell Co)*

Finned-tube sections are used for heating or cooling large volumes of gas at atmospheric pressure. The gas is blown across the finned surface, while either steam or some cooling medium flows through the tubes. For example, Fig. 72 shows a finned-tube section intended for use as a generator air cooler. The air flows from the cooler to the generator and back in a closed circuit and is cooled by cooling water flowing through the tubes of the cooler.

42 Pipe Coils —Pipe coils are an inexpensive type of heat-transfer equipment and are made in innumerable forms Two of the simpler forms are shown in Fig 73 Pipe coils are used
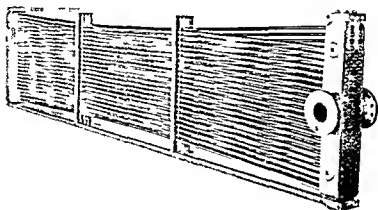
Fig 71 —Open type section with bowed tubes   *(Courtesy of the Griscom Russell Co)*

as gas or liquid coolers the coils being submerged in cooling water they are used as solution heaters the heating steam
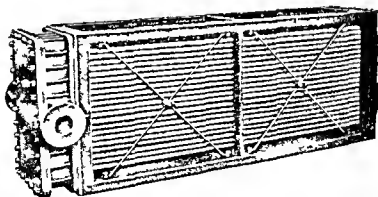
Fig 72 —Open type section with finned tubes   *(Courtesy of the Griscom-Russell Co)*

flowing through the coils and they are used as refrigerant evaporators the refrigerant evaporating as it flows through the coils

43. Hot-water Heaters.—The necessity of producing large quantities of hot water arises so frequently that several kinds

(a)

(b)
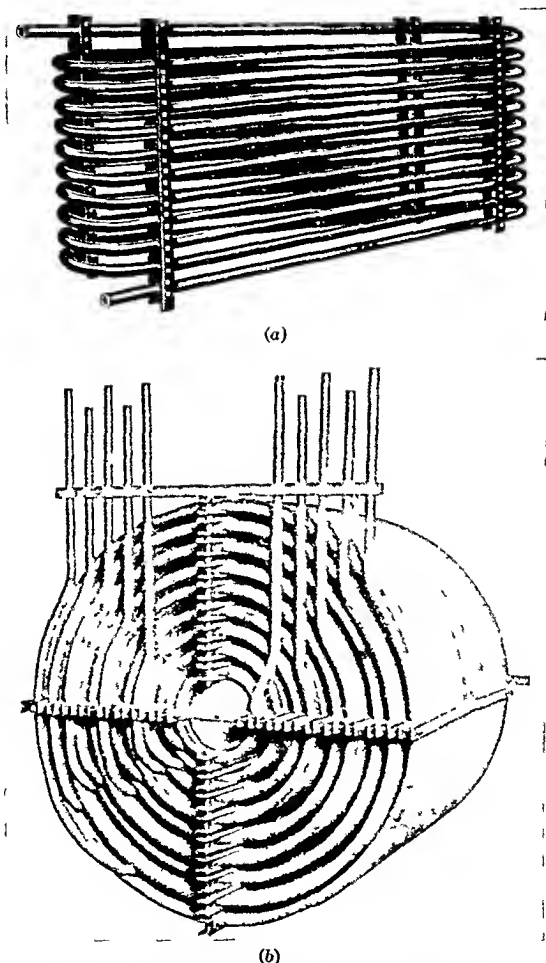
FIG. 73 —Typical pipe coils   (a) Oblong coil.   (b) Closely nested helical coils with all terminals at one end.   (*Courtesy of the Whitlock Coil Pipe Co.*)

of equipment designed especially for this purpose are manufactured. These heaters use steam as the heating medium and are of two types: storage heaters and instantaneous heaters.

Storage heaters are intended for use where large quantities of hot water are required at irregular intervals. They consist

(a)
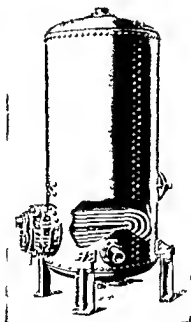
(b)

Fig 74 —Hot-water storage heaters (a) Horizontal type (b) Vertical type
(Courtesy of the Patterson-Kelley Co)

of a hot-water storage tank equipped with a bundle of U tubes, as shown in Fig 74 These heaters are sometimes made with

two separate tube bundles, one for high-pressure steam and the other for exhaust steam.

Instantaneous heaters differ from storage heaters in that the water to be heated flows through the tubes while the heating steam is in the shell. A typical instantaneous heater of the shell-and-tube type is shown in Fig. 75. Another design using coiled tubes is shown in Fig. 76. These heaters are particularly adapted for use where the demand for hot water is fairly steady.

44. Miscellaneous Types of Equipment. *a. Jacketed Vessels.* Jacketed vessels are frequently used for boiling liquids and are usually heated by steam condensing in the jacket. The heat-
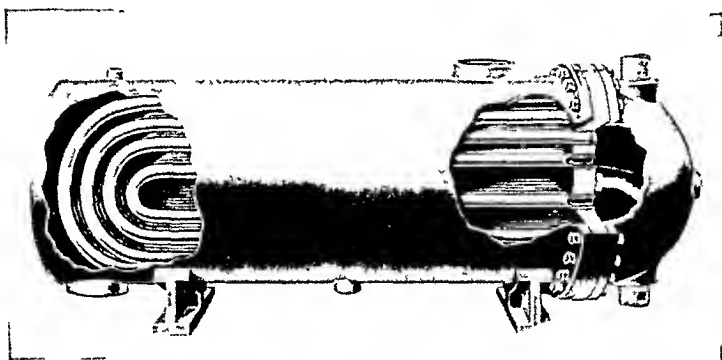
Fig. 75.—Instantaneous hot-water heater equipped with U tubes. (*Courtesy of the Patterson-Kelley Co.*)

transfer surface in contact with the liquid being boiled is relatively small but can be easily cleaned. For this reason jacketed vessels are used in place of submerged tube bundles for applications, such as cooking, that require frequent and thorough cleaning of all surfaces.

b. Submerged Steam Jets.—If the addition of some water is not objectionable, liquids can be heated by a submerged jet of steam. The rate of steam flow should be regulated so that all the steam is condensed and none escapes from the surface. By directing the jet properly, the steam can also be made to help circulate the liquid.

c. Sprays.—Gases can be very efficiently heated or cooled by a spray of liquid if saturating the gas with the liquid vapor is not objectionable. This system is sometimes used, for example,

in air conditioning installations The air to be cooled flows through sprays of cold water and, because of the large amount of water surface, is cooled very nearly to the temperature of the water

d Regenerative type Equipment—Regenerative-type heat exchangers differ from all the previously mentioned types in that the heat to be transferred is stored within the exchanger for a period of time These exchangers usually consist of a chamber filled with a checkerwork of firebrick, cast iron or other material capable of absorbing considerable heat, and they operate in a cyclic manner The hot fluid flows through the exchanger first and gives up heat to the checkerwork The cold fluid then flows through the exchanger and absorbs this heat Exchangers of this type are regularly used for preheating air for blast furnaces and open hearth furnaces

45 Tube Sizes and Materials The choice of tube size usually depends upon the service for which the equipment is to be used and upon economic considerations Smaller diameter tubes and closer spacing allow more surface to be installed in a given size tube bundle Also, for the same velocity of flow film coefficients in small tubes are higher than in large Larger tubes

Fig 76—Instantaneous hot-water heater equipped with coiled tubes *(Courtesy of the Grascom-Russell Co)*

are easier to clean however, and wider spacing allows cleaning lanes for mechanical cleaning of the outside surface Regarding tube-wall thickness thinner tubes are cheaper and are easier to expand into the tube sheets but they wear out faster Regarding tube lengths fewer tubes are required if longer tubes are used, and consequently cleaning and repairing are easier The

most frequently used tube sizes together with certain useful geometric data are listed in Table V of the Appendix.

Different materials vary in their ability to resist each of the various factors affecting the life of tubes. Since many combinations of these factors occur, tubes made of a number of different materials are necessary. Some of the more common tube materials are low-carbon steel, stainless steel, copper, brass, Admiralty, Muntz, copper-nickel alloys, aluminum, and lead.

46. Tube-installation Methods.—The commonest method of installing tubes consists of expanding the ends of the tubes into the tube-sheet holes by means of a rotating, roller expanding tool,
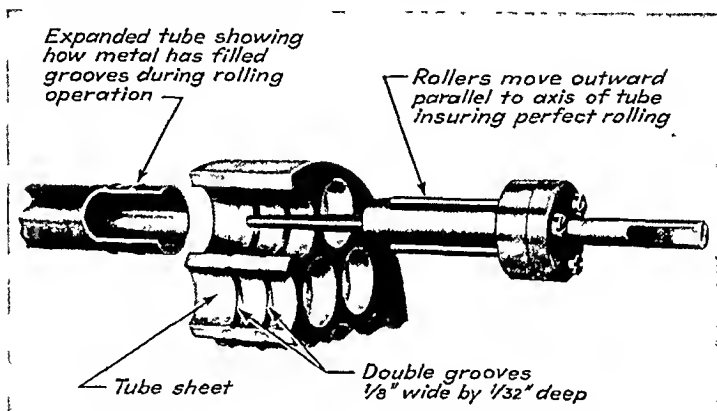
Fig. 77.—Method of expanding tubes into tube sheet. (*Courtesy of the Ross Heater and Mfg. Co.*)

as shown in Fig. 77. The tube-sheet holes are drilled undersize and are then reamed to a diameter slightly larger than the outside tube diameter. One or more grooves are frequently cut inside the holes in order to reduce the possibility of leakage and to increase the structural strength of the joints. Care must be taken not to overroll the tubes, since this would reduce the thickness of the tube walls at the point where corrosion and erosion are most severe. The inlet ends of the tubes are sometimes flared in order to reduce the likelihood of a contraction of the stream taking place. Expanding tubes by drifting (driving a conical pin into the ends of the tubes) is unsatisfactory and is seldom done.

Tubes are also frequently installed with ferrules, as shown in Fig 58 This method of installation permits easy tube replacements but has the disadvantage of involving a large number of joints to be kept tight

Tubes are sometimes sweated or soldered into the tube sheets. This method is particularly adapted to the installation of small-diameter tubes which would be difficult to expand It can also be used if the tubes are spaced so closely that expanding the
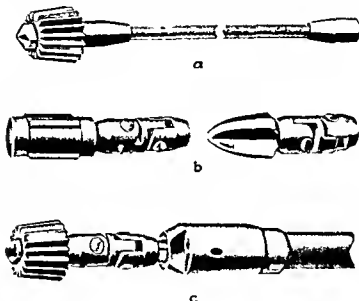
Fig 78 —Scale cutters for small tubes (a) Cone cutter with flexible connection for cleaning U tubes (b) Expanding-blade head and drill head (c) Cone cutter universal joint and steam or air-driven motor *(Cour sy of the Airetool Mfg Co)*

tubes might damage the tube sheet The method cannot be used if the equipment is to be used at high temperatures

Tubes can also be welded into the tube sheets but this method of installation makes tube replacements very difficult

47 Cleaning of Tubes—Cleaning the inside surfaces of tubes is facilitated by providing the headers with covers, such as those shown in Figs 48 55, and 72, that can be removed without breaking any of the pipe connections Removable plugs opposite the ends of each tube, as shown in Fig 71, also allow easy access

to the inside of the tubes. Cleaning the outside surfaces is facilitated by removable tube bundles and wide spacing of the tubes and baffles.

Hard, scaly deposits inside the tubes can be removed by means of cutters, such as those shown in Fig. 78. These cutters are driven by air-, steam-, or water-operated motors and are usually provided with a universal joint or some other type of flexible connection to permit their being used in bowed or U tubes. Soft or gummy deposits inside tubes can be removed by motor-driven revolving brushes or by rubber plugs forced through the tubes by water pressure.

Hard, scaly deposits on the outside surfaces of tubes can be removed by sandblasting, if the tube bundle is removable, or by motor-driven vibrators or knockers which are run through the inside of the tubes. Soft, gummy deposits can be removed with hand brushes. If the tube surfaces are not readily accessible, certain types of deposits can sometimes be dissolved by means of chemical cleaning fluids.

CHAPTER VII

INSULATION

48 Introduction—Any material having a low thermal conductivity can be used as insulation. A good insulating material must possess certain other properties, however. If it is to be used at high temperatures, it must be able to withstand these temperatures without deteriorating. If it is to be used at low temperatures, it must be either protected from or unaffected by moisture. For some applications the material may need to have sufficient structural strength to be self-supporting, it may need to be tough enough to resist rough usage and vibration, and it may need to be fireproof or resistant to chemical action. For other applications the material may need to be light in weight, odorless, and verminproof. In addition to these various factors, the choice of the kind of insulating material as well as the thickness to be used depends upon the cost. Other factors that must be considered in this connection are the ease of application, the salvage value of the insulation, and the cost of the heat or of the refrigeration that the insulation will save.

49 Basic Insulating Materials—Practically all industrial insulating materials are made from the following basic materials: asbestos, magnesium carbonate, diatomaceous silica, vermiculite, rock wool, glass wool, cork, cattle hair and wool.

Asbestos, a mineral possessing a crystalline, fibrous structure is obtained from asbestos mines in the form of fine, silky crystals. These crystals are easily separated by successive crushings into individual fibers. In order to be of commercial value, the asbestos must possess length and fineness of fiber combined with infusibility, toughness, and flexibility.

Magnesium carbonate is produced from dolomite limestone through a series of chemical processes including calcination of the dolomite, slaking in water, the addition of carbon dioxide to eliminate calcium carbonate, and precipitation of the magnesium carbonate. Magnesium carbonate owes its low con

ductivity to the great number of microscopic dead air cells that it contains.

Diatomaceous silica, also called *diatomaceous* or *infusorial earth,* consists of the siliceous skeletons of microscopic plants called *diatoms.* These diatoms live in water; and when they die, their skeletons fall to the bottom of the sea or lake and build up deposits of diatomaceous earth. Such deposits have been formed in the past and have been hardened into solid rock, the individual boxlike skeletons remaining unaltered, however. These earths are generally white or gray in color and are soft and friable.

Vermiculite is a micaceous mineral which possesses the peculiar characteristic of expanding in one direction only on being heated. The expansion takes place at right angles to the plane of cleavage and may amount to as much as sixteen times the original volume.

Rock wool and *glass wool* are made by melting silica minerals in a furnace and pouring the molten mass in fine streams into a high-velocity jet of steam. The molten mass is blown into silky fibers which rapidly cool and solidify. The resulting woollike formation is chemically inert and incombustible and has a low thermal conductivity because of the many void spaces between the fibers.

Cork is the outer layer of the bark of an evergreen oak tree known as the *cork oak.* This bark becomes unusually thick and is stripped off every 8 or 10 years. Cork is cellular in structure and owes its low thermal conductivity to the air entrapped within the cells.

50. Pipe Covering for Hot Lines.—*a.* Insulation for covering hot pipe lines is available in the form of half sections, whole sections, segmental blocks, blankets, and tape.

Half sections are the most commonly used form of insulation for pipe sizes up to about 10 in. They are held in place by metal bands or by loops of soft iron wire spaced from 9 to 12 in. apart. The ends of the wire are twisted, bent over, and hammered into the insulation so as to leave no projections. For extra-thick insulation, the half sections are applied in two layers, the circumferential and the longitudinal joints being staggered.

Whole sections can be used only when the insulation is sufficiently flexible to allow its being slipped over the pipe when split longitudinally along one side. This method of application is

advantageous in that it increases the effectiveness of the insulation by decreasing the number of joints

Flat blocks or sets of *curved segmental blocks* are used for insulating pipes larger than 10 in These blocks are held in place by soft iron wire loops and a thin coat of hard finish asbestos cement is applied over them to present a smooth even surface

When any of the foregoing types of rigid insulation are used the insulation is stopped off a sufficient distance from all flanges so that when the ends are beveled back at an angle of 45 deg the flange bolts may be easily removed

Blankets consisting of some insulating material held between two sheets of metal lath are applied by wrapping them around the pipe and lacing the edges of the metal lath together with wire

Insulating tape is sometimes used when the application of sectional insulation is impractical It is wrapped spirally around the pipe and held in place by wire clamps

b 85% *Magnesia* is the most commonly used kind of insulation for temperatures up to 600°F It has a low thermal conductivity is light in weight is easily cut and fitted is not affected by steam or water leakage and is sufficiently strong to withstand all ordinary usage It consists of a mixture of approximately 85 per cent magnesium carbonate and 15 per cent asbestos fiber The asbestos fiber is added to provide the necessary bonding and reinforcing qualities to allow the material to be molded into half sections and blocks as shown in Fig 79 The half sections are 3 ft long and vary in thickness from $\frac{7}{8}$ in to $1\frac{1}{4}$ in depending upon the pipe size The flat or curved segmental blocks are usually either 3 by 18 in or 6 by 36 in in size and are from $\frac{1}{2}$ to 4 in thick

The method of finishing depends upon the location On piping located indoors a canvas jacket may be sewed on over rosin sized paper or in the case of the half sections the flaps on the light canvas jacket which is regularly furnished as an integral part of the sections may be pasted down and the joints covered with brass lacquered bands On piping located outdoors the insulation may be covered with a jacket of roofing paper the joints all being lapped over and sealed with asphalt cement On horizontal pipes the seams of the jacket are placed at the side of the pipe with the lap turned down in order to shed water The jacket is secured in place by copper wire loops spaced about 4 in

apart. Where the insulation is in danger of mechanical injury, a metal jacket may be used.

c. Calcined diatomaceous silica and asbestos fiber, mixed with a bonding material, are used in the manufacture of an insulation that is suitable for temperatures up to 1900°F. The diatomace-
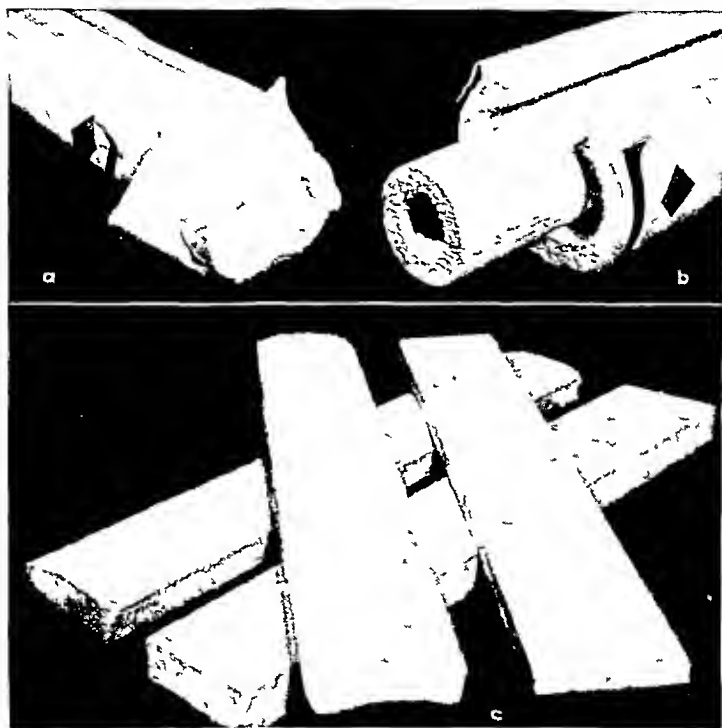
Fig 79 —85 % Magnesia insulation   (a) Single-layer pipe insulation   (b) Double-layer pipe insulation   (c) Blocks and lagging. (*Courtesy of the Keasbey & Mattison Co*)

ous silica is calcined, or burned, at a temperature of approximately 2200°F. before being mixed with the asbestos fiber, in order to limit the shrinkage of the finished product in service. This type of insulation does not have so low a thermal conductivity as 85% Magnesia but can be used at much higher temperatures. For this reason it is frequently used as an inner layer in combination with 85% Magnesia, as shown in Fig. 5 in Chap. I.

A sufficient thickness is used so that the temperature of the latter does not exceed 600°F the maximum allowable temperature for 85% Magnesia The greater insulating efficiency of the 85% Magnesia is thus combined with the higher heat resistance of the high temperature insulation to produce a covering that has a maximum efficiency for a given thickness of insulation This combination also reduces the cost of the insulation since 85% Magnesia is less expensive than the high temperature insulation

This type of insulation is molded into half sections 3 ft long from 1 to 3 in thick For pipe sizes from $\frac{1}{4}$ to 10 in For larger pipe sizes it is ordinarily furnished in segmental blocks

d Laminated asbestos felt insulation is especially suited to applications involving excessive vibration or rough usage It consists of approximately 40 layers per inch of thickness of a felt made of asbestos fiber and particles of finely ground sponge It has an unusually low thermal conductivity because of the great amount of dead air entrapped between the layers of felt and it possesses considerable toughness because of the felted nature of the material It is more expensive than molded insulations such as 85% Magnesia because it must be fabricated It may be used for temperatures up to 700°F

Sections for pipe covering are manufactured by winding the layers of felt one over the other and cementing them together at intervals The resulting cylinder is then split lengthwise to permit application For pipe sizes 3 in and larger it is split on one side only since the material is sufficiently flexible to permit the sections to be slipped on to the pipe as shown in Fig 80 For pipe sizes smaller than 3 in the cylinder is cut into half sections The sections are 3 ft in length and are available in thicknesses from 1 to 3 in For service indoors the sections are supplied with canvas jackets and are held in place with brass lacquered bands For service outdoors they are supplied with integral waterproof asbestos jackets

e Felted asbestos insulation is also suitable for applications involving vibration expansion and contraction It can be used for example for covering gas engine exhaust lines It is made from asbestos fibers felted and molded into half sections and can be used for temperatures up to 1000°F It is made in 3 ft sections from 1 to 3 in thick **to fit standard** pipe sizes up to 12 in

f. Felted rock wool secured between two layers of metal lath by tie wires, as shown in Fig. 81*a*, is used for temperatures up to 1000°F. Because of its flexibility, this type of insulation is shipped flat and is applied by wrapping it around the pipe and lacing the edges of the lath together with wire. For indoor work the insulation is coated with asbestos cement and finished with a
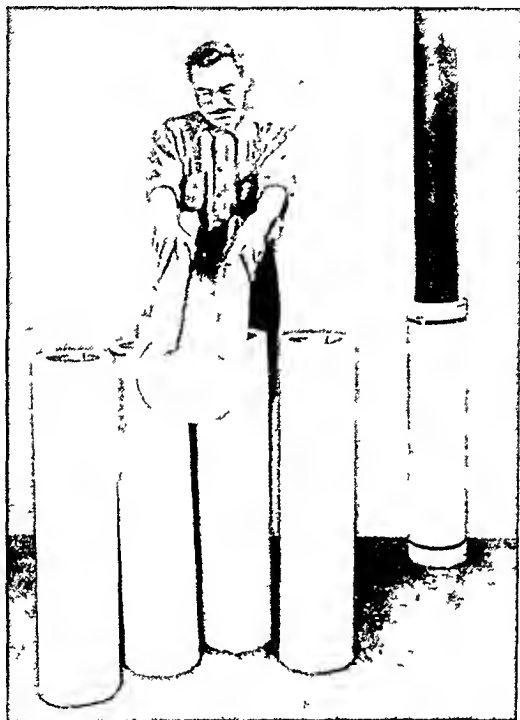
FIG. 80 —Laminated asbestos-felt pipe insulation. (*Courtesy of the Johns-Manville Co*)

canvas jacket. For outdoor work it is finished either with a coat of weatherproof cement or with a galvanized-iron outer casing, as shown in Fig. 81*b*. The latter is particularly suited to installations where a fireproof and weatherproof jacket is required or where the insulation must be frequently removed for repairs, changes, or inspection. This type of insulation is made in sections 2 ft. long, in thicknesses from 1 to 4 in., for pipe sizes

2 in in diameter and larger   Special shapes for flanges and
bends can be obtained

g Glass wool supported by straight-line wire netting shown
in Fig 82 is used for covering pipes 8 in in diameter and larger
at temperatures up to 1000°F   Since the wire netting is on the
outside face only this type of insulation is particularly adapted
for covering spirally riveted or spirally welded pipe   Rods
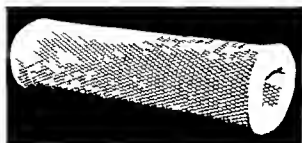
(a)

(b)

Fig 81 —Felted rock wool pipe insulation   (a) Secured between metal lath
(b) Equipped with galvanized iron outer casing   (*Courtesy of the Johns Manville
Co*)

bound to the edges of the wire netting form a selvage which
facilitates lacing the covering together   The sections are made
2 ft long in thicknesses from 1 to 4 in

h Quilted asbestos blankets shown in Fig 83 are made of
asbestos cloth filled with loose asbestos fiber and are used on
pipes where frequent inspection requiring the removal of the
insulation is necessary   These blankets are easier to apply than
molded insulation and are not subject to breakage   They are
provided with metal hooks and are laced on to the pipe with

Fig. 82.—Glass-wool pipe insulation. *(Courtesy of the Owens-Corning Fiberglas Corp.)*

Fig. 83.—Quilted asbestos blankets. *(Courtesy of the Johns-Manville Co.)*

wire They may be used for temperatures up to 900°F and
are manufactured in sections 3 ft long in thicknesses from 1½ to
3 in.

1 *Insulating tape*, consisting of loosely twisted strands of
asbestos cloth can sometimes be used where lack of space or
numerous bends make the application of rigid, sectional insula

Fig. 84.—Insulating tape (*Courtesy of the Johns-Manville Co.*)
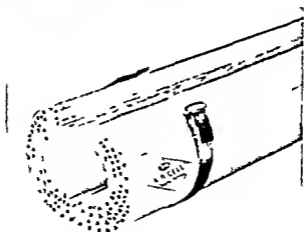
Fig. 85.—Corrugated asbestos pipe insulation *Courtesy of the Keasbey &*
Mattison Co.)

tion impractical The tape can be used for temperatures up to
400°F and will withstand considerable vibration and rough
usage. It is wrapped spirally around the pipe and is held in
place by wire clamps as shown in Fig. 84 It is made in 50-ft.
rolls, in width up to 2⅝ in, and in thicknesses up to ⅜ in

2 *Corrugated asbestos* insulation shown in Fig. 85 may be
used for temperatures up to 300°F It is an inexpensive type of

insulation and is regularly used for covering low-pressure steam-heating lines. It consists of alternate layers of plain and corrugated asbestos paper, cemented together at the points of contact, and is made in half sections which are hinged together by either a canvas or a heavy asbestos-paper jacket. The sections are held in place on the pipe by pasting down the flap on the canvas jacket or by inserting metal staples into the paper jacket. The circumferential joints are covered with metal bands.

This type of insulation is made in sections 3 ft. long, in thicknesses from two to eight plies, to fit all standard pipe sizes. The standard ply is $\frac{1}{4}$ in. thick, although finer corrugations averaging $\frac{1}{6}$ in. per ply are also made.
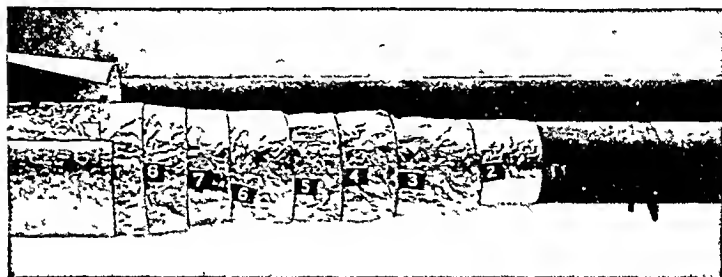
Fig. 86.—Aluminum-foil pipe insulation. (*Courtesy of the Alfol Insulation Co*)

k. Wool-felt insulation, designed particularly for use on either hot- or cold-service water lines, may be used for temperatures up to 225°F. It consists of an inner liner of waterproofed asbestos felt covered with a continuous wrapping of creped-wool felt, a wrapping of asbestos paper over the wool felt, and a canvas jacket pasted over the asbestos paper. The asbestos paper is added to make the insulation match other insulation in appearance, wool felt itself being gray in color and darker than either asbestos or magnesia. The insulation is split longitudinally to permit application, the canvas jacket and outer layer of felt acting as a hinge. The sections are held in place by pasting down the canvas flap, the joints between the sections being covered with metal bands. This type of insulation is manufactured in sections 3 ft. long, in thicknesses of $\frac{1}{2}$, $\frac{3}{4}$, and 1 in. for all pipe sizes from $\frac{1}{2}$ in. to 5 in.

l. A comparatively recent development is the use of *crumpled aluminum foil* for insulating pipes. This type of insulation has

a very low thermal conductivity, is unusually light in weight, and has a very low heat storage capacity The aluminum foil used is approximately 0 0003 in thick and is furnished in smooth rolls It is crumpled by hand before being wrapped around the pipe and is applied loosely enough that the air spaces between the sheets are about $\frac{3}{8}$ in thick Since the foil has no structural strength itself, it must be protected from crushing or injury by a covering of rigid material such as galvanized iron or pressed asbestos, as shown in Fig 86 Note the wire frames used to support the galvanized iron cover The aluminum foil will
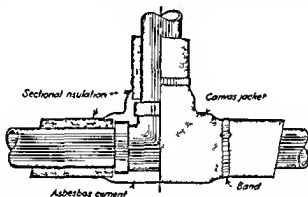
Fig 87 —Asbestos cement insulation for small fittings and valves *(Courtesy of the Johns-Manville Co)*

withstand considerable vibration and can be used for temperatures up to 1000°F

51 Valves and Fittings on Hot Lines—Valves and fittings on hot lines in the smaller pipe sizes (3½ in or smaller) are usually insulated with hard finish asbestos cement as shown in Fig 87 Successive layers of cement, are applied until the total thickness is equal to that of the insulation on the adjacent piping Each layer is allowed to dry thoroughly before the next layer is applied the final layer being troweled to a smooth, hard finish

On the larger pipe sizes (4 in or larger) the bodies of flanged fittings and the entire surface of screwed fittings are covered with block insulation as shown in Fig 88 The blocks are usually made of the same material as that used on the adjacent piping and are held in place by loops of soft iron wire They are applied to a thickness ½ in less than that of the insulation

on the piping, and hard-finish asbestos cement is applied over them to make the total thickness of insulation equal to that on the piping.
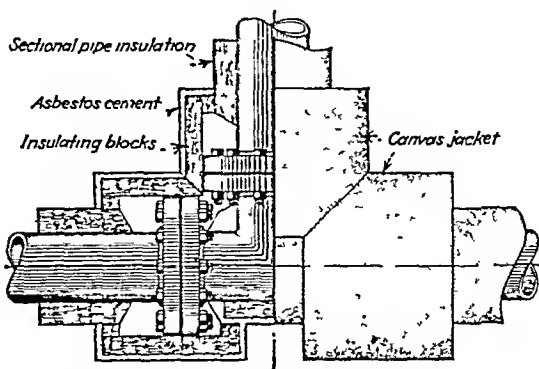
FIG 88 —Block insulation for large fittings and valves   (*Courtesy of the Johns-Manville Co*)

Flanges may be insulated in either of two ways: If the insulation is to be permanent, blocks long enough to extend about 2 in. over the adjacent pipe insulation are wired around the
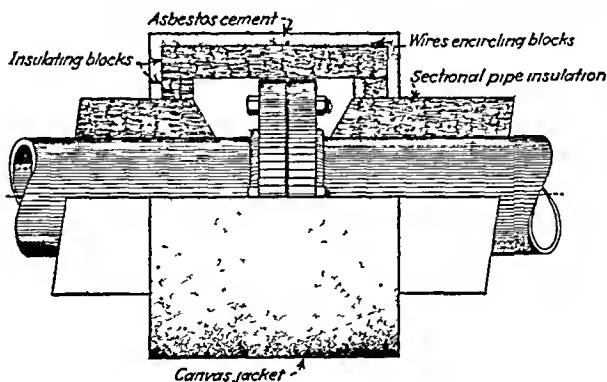
FIG. 89.—Permanent type of insulation for flanges   (*Courtesy of the Johns-Manville Co*)

flange and are finished with a coating of asbestos cement, as shown in Fig. 89. If the insulation is to be removable and replaceable, either sectional or block insulation can be used to

construct a split cover   If sectional insulation is used the half
sections must be large enough to encircle the flange completely
and long enough to extend about 2 in over the pipe insulation
as shown in Fig 90   The sections are held in place with wire,
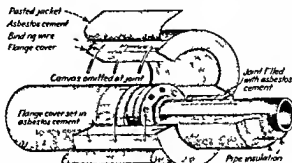
Fig 90 —Removal and replaceable sectional insulation for flanges   (Courtesy of
the Johns Manville Co)

and the joint between the flange insulation and the pipe insula
tion is sealed with asbestos cement   A smooth even surface is
obtained by applying a thin coat of cement over the entire cover
If the cover is made of blocks the blocks are wired to a frame
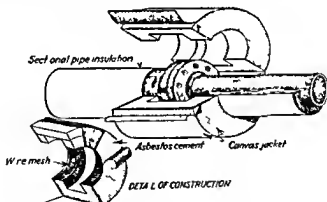
Fig 91 —Removable and replaceable block insulation for flanges   (Courtesy of
the Johns Manville Co)

made of $\frac{1}{2}$ in square mesh galvanized wire cloth as shown
in Fig 91   The mesh is split at the ends and bent down to
inclose asbestos cement used to fill the annular space between
the blocks and the pipe insulation   The cover is wired in place

on the flange, and the entire surface is covered with a coat of asbestos cement.

The insulation on valves and fittings located indoors is finished with a covering of canvas, cut and stretched to cover the entire

(a)

(b)

(c)   (d)
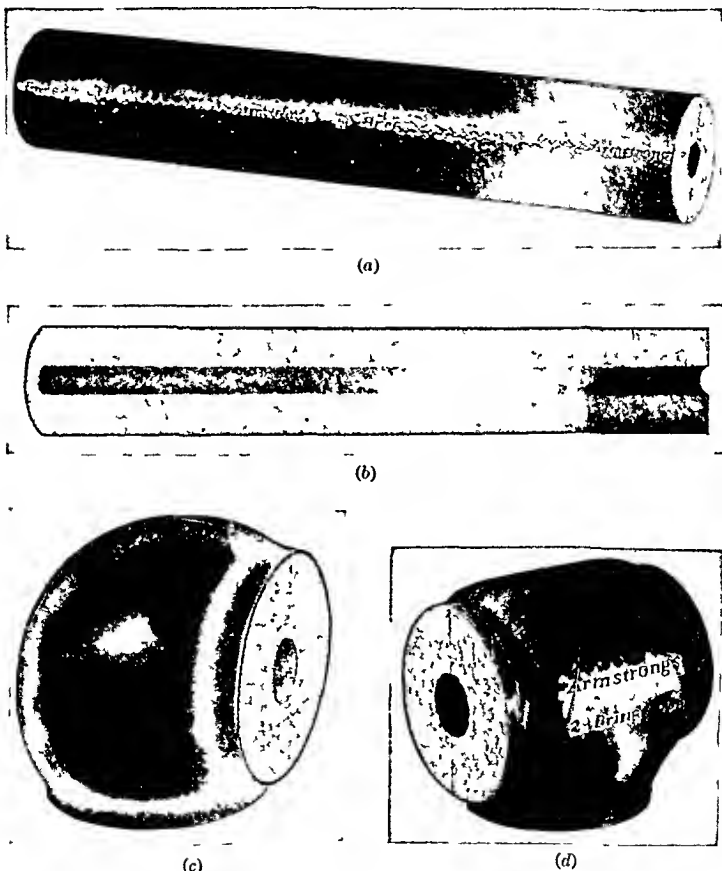
Fig 92 —Typical sections of cork insulation. (a) Full section of pipe covering. (b) Half section of pipe covering. (c) Screwed ell cover. (d) Screwed tee cover. (*Courtesy of the Armstrong Cork Co.*)

surface and held in place by paste. If the insulation is exposed to the weather, it is finished by applying a $\frac{1}{4}$-in. coat of cement consisting of two parts asbestos fiber and one part Portland

cement   This is covered in turn with a $\frac{1}{4}$ in coat of weather
proof asphalt cement

52 Pipe Covering for Cold Lines—*a* In order to be per
manently effective, the insulation on all pipes colder than the
surrounding atmosphere must be finished with a surface that is
practically airtight   Otherwise, the insulation will eventually
become completely water soaked by the moisture condensed out
of the air filtering into the insulation   This will not only destroy
the effectiveness of the insulation but on freezing may burst and
damage it

b Cork insulation is manufactured from granules of pure
cork compressed into molds under high pressure and baked   The
baking process liquefies the natural resin in the cork, which
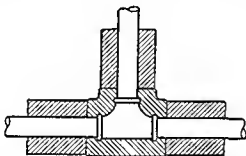
Fig 93 —Cross section of cork covering for screwed tee

cements all the particles into a homogeneous mass   The inside
surface of the insulation is machined to form a snug fit on the
pipe and the outside surface is finished with a heavy mastic
coating   Sections are molded to fit practically all standard
screwed and flanged fittings as well as to fit straight pipe   Some
typical sections are shown in Fig 92

The insulation is held in place by loops of copper clad steel
wire   all the joints being coated with waterproof cement as the
insulation is applied   Screwed fittings are usually covered first
The sections for covering these are machined to form a snug fit
as shown in Fig 93   Straight pipes are covered next the sec
tions being cut long enough to be wedged slightly against the
covers on the screwed fittings or against the bolt heads of the
flanged fittings   Flanged fittings are covered last   The sec
tions for covering these do not fit snugly on to the fitting but
rest on the adjacent pipe insulation as shown in Fig 94   The

annular space between the fitting and the sectional covering -is filled with a mixture of melted paraffin and granulated cork. This mixture is poured through a hole bored through the sectional covering at the highest point and is allowed to solidify.   On indoor lines the insulation is usually finished with a coat of asphalt paint; on outdoor lines it is covered with a layer of roofing paper.

Cork insulation is made in three thicknesses known, respectively, as *Special Thick Brine*, *Brine Thickness*, and *Ice Water Thickness*.   *Special Thick Brine* varies in thickness from 2.63 to 4.00 in., depending upon the pipe size, and is intended for lines
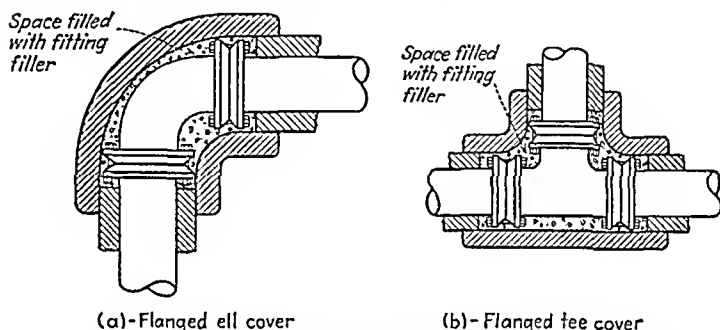
(a)-Flanged ell cover   (b)- Flanged tee cover
Fig. 94.—Cross section of cork covering for flanged fittings.   (*Courtesy of the Armstrong Cork Co.*)

carrying fluids at temperatures between 0 and −35°F.   For temperatures below −35°F., the thickness of the insulation is increased by applying segmental blocks (lagging) over the sectional covering.   *Brine Thickness* varies from 1.70 to 3.00 in. and is intended for temperatures between 0 and 35°F.   *Ice Water Thickness* varies from 1.20 to 1.93 in. and is intended for temperatures above 35°F.   All three thicknesses are made in sections 3 ft. long to fit standard pipe sizes up to 8 in.   The insulation is made in the form of segmental blocks for larger pipe sizes.

e. Loose *rock wool* impregnated with a waterproof binder is molded into half sections and segmental blocks for covering straight pipe.   The half sections are made with an integral waterproof jacket which acts as a hinge, as shown in Fig. 95. The sections are held in place by cementing down the flap with waterproof cement and twisting several loops of wire around each

section The ends of the sections are sealed against the infiltration of air by a coating of waterproof cement, applied as the sections are slipped into place, and by wrapping the joints with waterproof tape

The sections are made in 3 ft lengths in *Heavy Brine*, *Brine*, and *Ice Water* thicknesses The *Heavy Brine* thickness varies from 2 82 to 4 00 in , depending upon the pipe size, and is intended for covering pipes carrying fluids at temperatures between 0 and −30°F It is made in half sections for pipe sizes up to 6 in and in segmental blocks for larger sizes For tem

FIG 95—Method of applying molded rock wool pipe covering   *(Courtesy of the Johns Manville Co)*

peratures below −30°F the thickness of the insulation is increased by applying segmental blocks over the *Heavy Brine* covering *Brine* thickness varies from 1 97 to 3 19 in and is intended for temperatures between 0 and 30°F It is made in half sections for pipe sizes up to 8 in and in segments for larger sizes *Ice Water* thickness varies from 1 4 to 2 0 in and is intended for temperatures above 30°F It is made in half sections for pipe sizes up to 10 in and in segments for larger sizes

Valves and fittings are insulated with loosely felted rock wool impregnated with asphalt This material is made in rolls 18 in wide with a nominal thickness of approximately 1½ in   The

felt is applied by cutting a piece from the roll two or three times the circumference of the fitting, folding it double or triple according to the thickness desired, and placing it around the fitting. It is secured in place with jute twine and sealed against the infiltration of moisture by a wrapping of waterproof tape, as shown in Fig. 96. On the larger sizes a second and third layer of the felt are applied, each layer being sealed with tape.

Fig 96 —Method of insulating valves and fittings with rock wool (*Courtesy of the Johns-Manville Co*)

The insulation is finished with a coat of waterproof cement troweled on over the tape.

d. Hair felt is made from cattle hair, chemically cleaned and felted. It is made in rolls (see Fig. 97) 50 ft. long, in widths from 3 to 9 ft., and in thicknesses from $\frac{1}{4}$ to 2 in. It is a generally useful material for low-temperature work.

Hair felt can be used in place of either cork or rock wool for covering refrigerant lines by building up a suitable number of layers of the felt. The required length of felt for each layer is wrapped around the pipe and held in place by twine. Water-

proof tape is wrapped spirally around the felt and waterproof cement is applied over the tape The insulation is finished with a canvas jacket sewed on over a layer of rosin-sized sheathing paper This type of covering requires considerable skill and care in its application and consequently is not commonly used.

Water pipes are frequently protected from freezing by a covering made of hair felt For severe conditions a suitable number of layers of the felt (see Sec 63) are bound on to the pipe with twine and covered with a canvas or a weatherproof jacket Precautions against the infiltration of air are not needed, since the pipe is warmer than the surrounding air, and therefore

Fig 97 —Roll of hair felt   *Court sy of the Johns-Manville Co)*

condensation of moisture in the insulation cannot take place If the pipes are exposed only to moderately low temperatures, or if the time that the surrounding air is below the freezing point is of short duration a covering designed especially for these conditions can be used This covering shown in Fig 98, consists of two thicknesses of hair felt protected on both the inside and outside faces by several layers of waterproofed wool felt. The sections are split longitudinally to permit application, the outer layer of wool felt acting as a hinge They are held in place by pasting down the flap on the canvas jacket and covering the joints with metal bands The sections are made in 3-ft lengths to fit all pipe sizes from ½ in. up but in only one thickness approximately equal to 1¼ in

Hair felt is also used for covering valves and fittings on lines insulated with either hair or wool felt. The felt is held in place with twine and is usually sealed against moisture either with a wrapping of waterproof tape and a coat of asphaltic paint or with a ¼-in. layer of asbestos cement.

e. Wool felt is used in the construction of two types of insulation intended for covering cold-water pipes. The first type, described in Sec. 50*k*, is made of creped-wool felt. The second
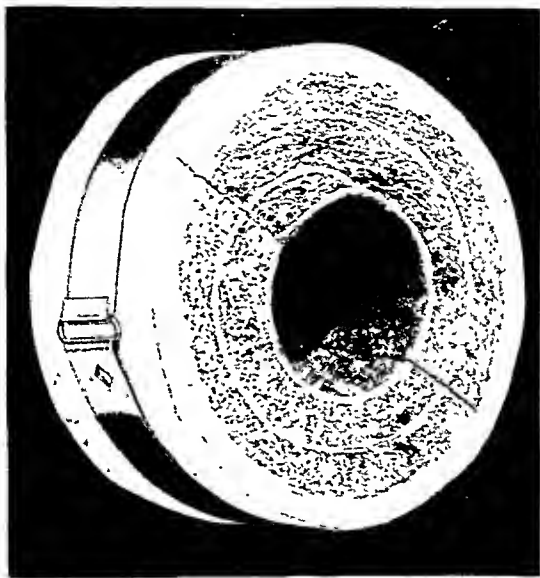
Fig. 98.—Hair-felt insulation for protecting water pipes from freezing. (*Courtesy of the Johns-Manville Co.*)

type, shown in Fig. 99, is intended for more severe operating conditions and is made of flat-wool felt protected from moisture by layers of waterproofed felt. It is made in single-layer construction for thicknesses up to ¾ in. and in double-layer construction for thicknesses up to 2 in. The sections are 3 ft. long and are made to fit standard pipe sizes up to 6 in.

These two types of insulation serve not only to keep the water in the pipes cold but also to prevent condensation on the outside of the pipe. The prevention of condensation is of particular importance in industries where dripping from pipes might

interfere with the manufacturing process or might damage finished goods. The thickness of insulation required (see Sec. 64) depends upon the temperature and relative humidity of the air surrounding the pipe and upon the temperature of the water in the pipe, and it must be such that the temperature of the outside surface of the insulation is above the dew point of the air.

53. Block Insulation for High Temperatures.—*a.* Large surfaces, both flat and curved, can be covered with insulation in
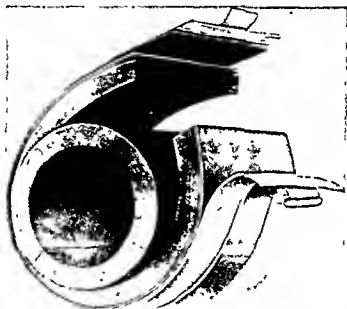
Fig. 99.—Wool-felt insulation for covering cold-water pipes.   (*Courtesy of the Johns-Manville Co.*)

the form of blocks. These blocks are made of the same materials that are used for making pipe covering. Blocks made of 85% *Magnesia* are the most commonly used and are suitable for temperatures up to 600°F. For temperatures up to 1900°F., blocks made of *calcined diatomaceous silica* and *asbestos fiber* are used. Both these kinds of blocks are made either flat or curved to any desired radius, in widths of 3, 6, or 12 by 18 or 36 in. long, in thicknesses from 1 to 4 in.

Blocks made of *laminated asbestos felt* can be used for applications involving excessive vibration or rough usage. They can

be used for temperatures up to 700°F and are made in widths
from 6 to 36 by 36 in. long, in thicknesses from $\frac{1}{2}$ to 4 in. They
are made flat but are sufficiently flexible to permit their being
applied to curved surfaces. Blocks made of *felted asbestos* can
also be used where the insulation is subjected to vibration or to
compression and strain. These blocks are 6 in. wide by 36 in.
long and from $\frac{1}{2}$ to 4 in. thick. They can be used for tem-
peratures from 500 to 900°F , depending upon how well they are
supported
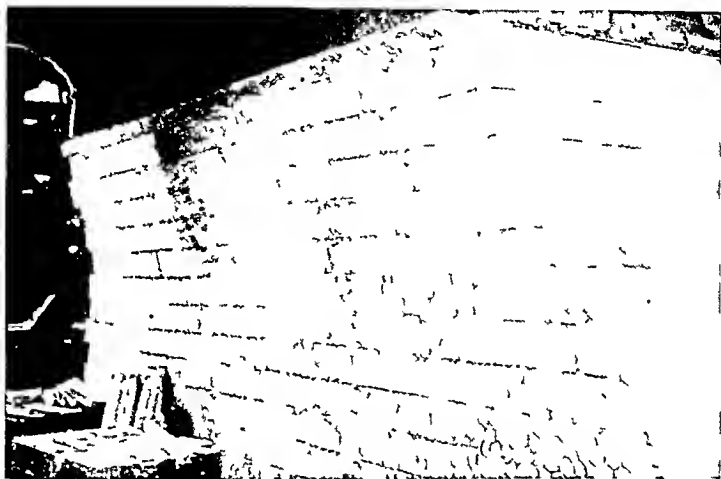
Fig 100.—85 % Magnesia blocks wired to flue and ready for final coat of asbestos
cement   (*Courtesy of the Johns-Manville Co*)

b. The usual method of applying block insulation is illustrated
in Fig. 100. It consists in wiring the blocks on to the surface,
stretching either hexagonal-mesh wire netting or expanded metal
lath over the blocks, and applying one or more layers of asbestos
cement over this   The blocks may be held temporarily in place
during erection by an *adhesive cement* (see Sec. 58).

One method of wiring the blocks on to flat metal surfaces is
shown in Fig. 101. Drilled angle clips are spot-welded to the
metal surface, and the tie wires are threaded through them   If
the blocks are to be applied to flat wooden surfaces, strips of
wood the same thickness as the blocks are nailed to the wall,
and the tie wires are stapled to these

The method of anchoring the tie wires used to hold block insulation in place on tanks drums and towers is illustrated in Figs 102 and 103   The blocks are held in place on the sides
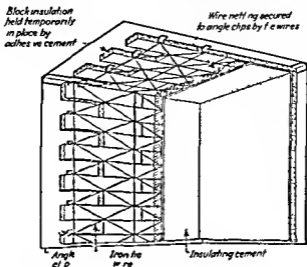
Fig 101 —Method of wiring block insulation on to flat metal surfaces   *(Courtesy of the Johns-Manville Co)*
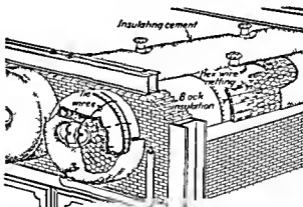
Fig 102 —Method of insulating drums and drum heads   *(Courtesy of the Johns Manville Co)*

of the vessels by loops of wire which pass completely around the insulation   If the blocks are not to encircle the vessels completely, angle irons are welded along the sides of the vessels where the insulation is to terminate and the tie wires are fastened

on to these. The tie wires for holding the blocks in place over the ends of the vessels are anchored on to pencil-rod rings which are fastened around the rims of the vessels and around any pipe connections or manhole covers, as shown. These rings are held in place by clips spot-welded on to the vessels. The insulation
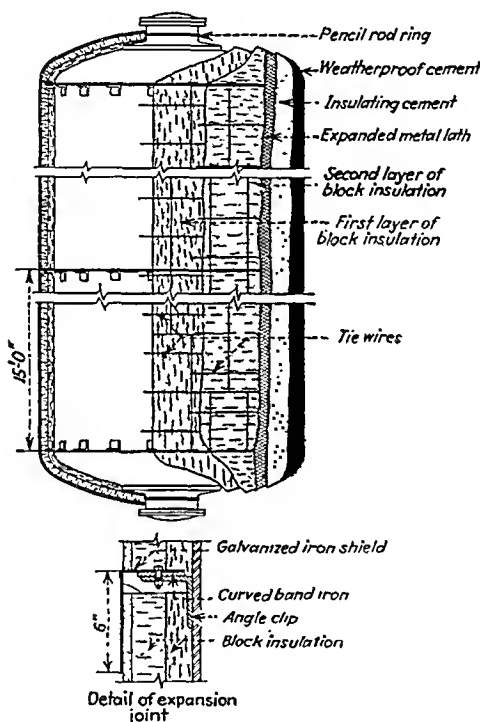
FIG. 103.—Method of insulating vertical tanks and towers. (*Courtesy of the Johns-Manville Co.*)

on vertical tanks and towers is supported at intervals of not more than 15 ft. by horizontal band-iron rings bolted on to angle clips, as shown in Fig. 103. These rings permit the inclusion of expansion joints and also serve as anchors for the tie wires.

In place of tie wires and wire netting, insulating blocks can be supported by some sort of rigid casing. Figure 104 illustrates this method of support applied to the insulation of a steam-boiler waterwall. Insulating cement is first applied to a thickness just

sufficient to provide a flat surface, and the blocks are held in place on this surface by the casing In other applications the shell is erected first, and the insulation is built up inside it

In special cases where the temperatures do not exceed 500°F and where the application of tie wires or other means of support is impractical, insulating blocks may be held in place by adhesive cement alone
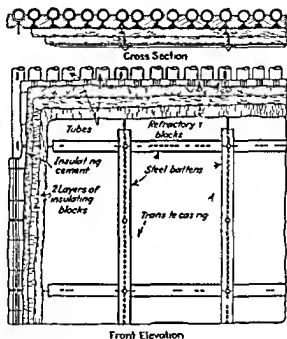
Fig 104   Insulating blocks supported by a rigid casing   (Courtesy of the Johns-Manville Co)

54 Block Insulation for Low Temperatures —a large surfaces subjected to temperatures below atmospheric can be covered with blocks made either of cork or of rock wool impregnated with a waterproof binder These blocks are manufactured by the same processes used in making the corresponding types of sectional pipe covering (Secs 52b and 52c) Flat surfaces are covered with relatively wide blocks and cylindrical surfaces are covered with narrow blocks commonly known as lags These lags have beveled edges to permit them to fit snugly Flat disks are made for covering the ends of tanks

Cork blocks for covering flat surfaces are made in widths from 12 to 36 in., in 36-in. lengths, and in thicknesses from 1 to 6 in. For ordinary service they are made without any special coating, but for exceptionally severe conditions they are made with an asphalt mastic coating ironed on under pressure. Cork lags are made 36 in. long, from 1½ to 6 in. thick, and are coated on both sides with a mastic finish. Disks are made from 1½ to 6 in. thick and any diameter desired. *Rock-wool* blocks are made

Fig. 105.—Showing the application of three layers of cork blocks to the walls and floor of an ice cream hardening room. (*Courtesy of the Armstrong Cork Co.*)

18 in. wide by 36 in. long and from 1 to 4 in. thick. Rock-wool lags are made 18 in. long, from 1½ to 4 in. thick, and from 2 to 5 in. wide, the width depending upon the diameter of the cylindrical surface to be covered. Disks are made from 1½ to 4 in. thick.

b. The blocks may be fastened to brick, stone, or concrete walls by setting them in either Portland-cement plaster or hot asphalt. Asphalt has the advantage of being waterproof but cannot be used on rough walls unless a leveling coat of Portland-cement plaster is first applied. The blocks are usually held in

place on floors and roofs by means of a mopping of hot asphalt
They are fastened to frame structures and other wooden surfaces
by nailing and they are fastened to flat metal surfaces such as
rectangular ducts, by means of either hot asphalt or waterproof
cement If two or more thicknesses of blocks are to be applied
to any surface, each additional layer can be fastened to the
preceding one by means of hot asphalt reinforced with wood
skewers As shown in Fig 105, the successive layers of blocks
are staggered, so that the joints between the blocks in any one
layer are covered by the blocks in the adjacent layers
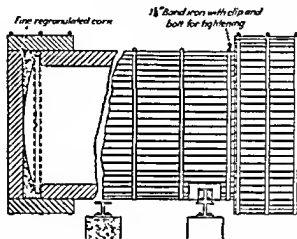
Fig 106 —Method of applying cork lagging to cylindrical tanks. *(Courtesy of the Armstrong Cork Co)*

Figure 106 illustrates the method used in applying cork lagging
to cylindrical tanks The edges of the lags are coated with
waterproof cement as they are applied A rope and spring being
used to hold them in place during erection The lags are clamped
permanently in place by No 16-gauge galvanized iron bands
1¼ in wide, spaced approximately 12 in apart The ends of
the tanks are covered with disks and the spaces between the
disks and the tank are filled with a mixture of granulated cork
and paraffin Rock wool lagging is applied in a similar manner
except that either loose or granulated rock wool is used to fill
the irregular spaces at the ends of the tank Horizontal tanks
should preferably be supported on saddles outside the insulation

The saddles should be large enough to support the tank without compressing the insulation

Low-temperature block insulation may be finished by painting, applying hot asphalt, or applying a coat of Portland-cement plaster. The completed insulation on two brine coolers and the piping connected to them is shown in Fig. 107. The insulation is finished with a coat of aluminum paint.
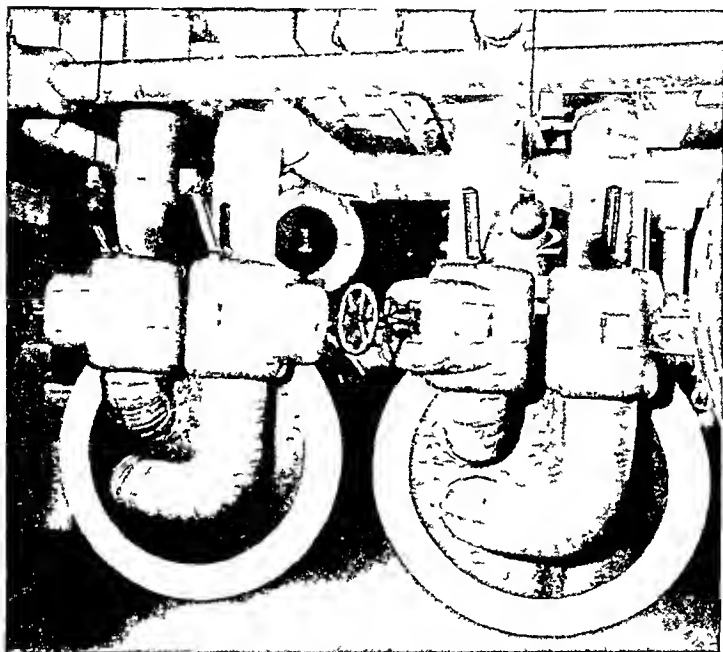
Fig 107.—Cork insulation on two brine coolers and the connected piping
(*Courtesy of the Armstrong Cork Co*)

55. Sheets.—Sheet insulation is made in a variety of forms. The simplest is *asbestos paper*, which is made either plain or corrugated. Asbestos paper is used for insulating furnace pipes, for protecting inflammable surfaces from heat, and as a fire retarder. *Asbestos millboard* is also used as a protection against fire, heat, or acid fumes and is frequently used as a fireproof lining for floors and partitions   It is made in sheets 42 by 48 in. and in thicknesses from $\frac{1}{32}$ to $\frac{3}{4}$ in. Various grades are made,

some suitable for temperatures as high as 1800°F , though the most commonly used grade is not generally recommended for temperatures above 400°F

Another type of sheet insulation, similar to but less dense than asbestos millboard, is made from *felted asbestos fiber* formed into sheets 24 by 36 in in size and from ½ to 4 in thick   This material is particularly suitable for use where the insulation is subject to vibration or to strain due to expansion and contraction, such as in boiler-tube doors   It can be used for temperatures from 500 to 900°F , depending upon how well it is supported
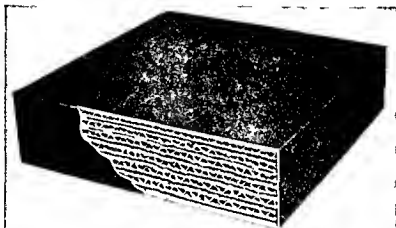
Fig 108 —Corrugated asbestos sheets protected by a refractory coating   (Courtesy of the Johns Manville Co)

An inexpensive type of sheet insulation suitable for temperatures up to 300°F , is made from alternate layers of *plain and corrugated asbestos paper* cemented together at the points of contact   These sheets are used for covering low pressure boilers, dry rooms warm air ducts and other surfaces whose temperatures are not extreme   The sheets are made in widths from 6 to 36 in , in 36- and 72-in lengths, and in thicknesses from ½ to 4 in   These sheets are also made with a protective refractory coating on the outside surface as shown in Fig 108   This coating allows them to withstand temperatures up to 500°F and makes them suitable for building drying ovens, japanning ovens, and other similar heated inclosures   If the sheets are vitrified throughout, they can be used at temperatures up to

700°F. This type of sheet is made both flat and curved, as shown in Fig. 109, and is used for lining steel stacks and flues.
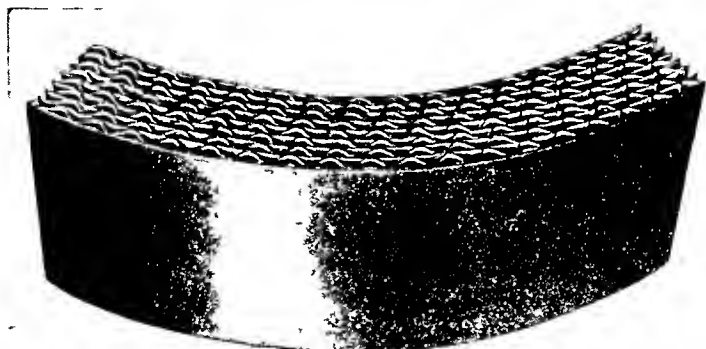
Fig 109.—Corrugated asbestos sheets vitrified throughout (*Courtesy of the Johns-Manville Co.*)

A sheet insulation intended particularly for use in railroad passenger cars is made of one or more plies of *hair felt* approximately ¼ in. thick. The individual plies are faced with kraft paper and are stitched together on approximately 5-in. centers. The assembled sheet consists of the desired number of plies inclosed in a layer of asbestos paper and a layer of muslin gauze and held together by stitching between the stitching on the individual plies. This type of insulation is made in widths from 36 to 54 in., in thicknesses from one to four plies, and in approximately 100-ft. lengths.

Fig. 110.—Sectional view of an aluminum-foil panel (*Courtesy of the Alfol Insulation Co*)

Sheet insulation for use in refrigerator cars is also made from *hair felt*, but waterproof paper is used in place of the kraft paper, and the insulation consists of a single ply. It is made in any width up to 108 in., in thicknesses from ½ to 2 in., and in practically any length desired.

Sheets of *aluminum foil* are used both as high- and as low-temperature insulation. The foil, approximately 0 0003 in. thick, is furnished in smooth rolls and is crumpled by hand before

being applied   The low thermal conductivity of this insulation
is due largely to the air spaces between the sheets the optimum
thickness of air space being about $\frac{3}{8}$ in   Since the foil has no
structural strength itself it must be protected from injury by a
suitable jacket   When used as low temperature insulation it
must in addition be protected from the infiltration of air
or moisture   A thin coat of lacquer is sometimes applied to the
foil to protect it from corrosion   Outstanding features of this

Fig 111 — All metal aluminum foil panels   *(Courtesy of the Alfol Insulation Co)*

insulation are its extremely light weight (approximately 3 oz
per cu ft) and its low heat-storage capacity

Panels made of sheets of smooth aluminum foil separated by
asbestos strips illustrated in Fig 110 are used for a number of
purposes   All metal panels illustrated in Fig 111 consist of
an outer sheet metal jacket an expanded metal lath basket
welded to this jacket and a filling of crumpled aluminum foil
sheets   These panels are suitable for covering tanks frac-
tionating towers and other large flat or curved surfaces   They
may be used for temperatures up to 1000°F
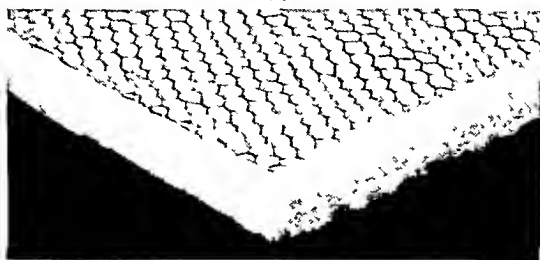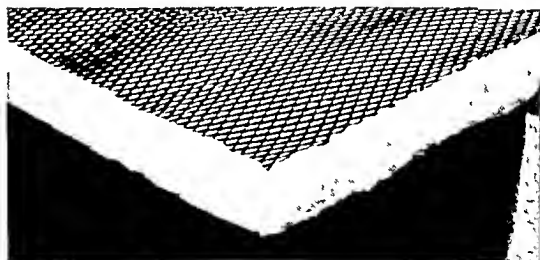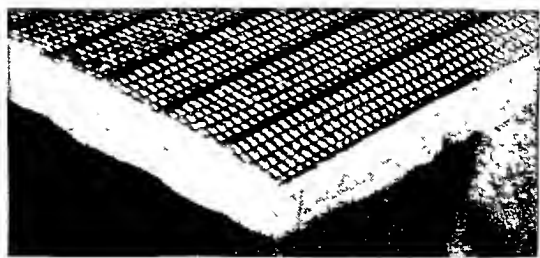
FIG. 112.—Glass-wool insulating blankets. (a) Metal facing on one side only.
(b) Facing of hexagonal-mesh wire netting. (c) Expanded metal-lath facing.
(d) Rib-lath facing. (*Courtesy of the Owens-Corning Fiberglas Corp.*)

56 Blankets—Large flat or curved surfaces such as those on ovens oil refinery equipment boilers breachings, and hot air ducts are frequently covered with insulating blankets The size of these blankets is more than five times the size practical for pieces of molded insulation, so fewer joints are required and faster application is possible In addition the edges of the blankets knit together so that there are no loose joints

Insulating blankets may consist of felted *rock wool* or *glass wool* secured between various fabrics as shown in Fig 112
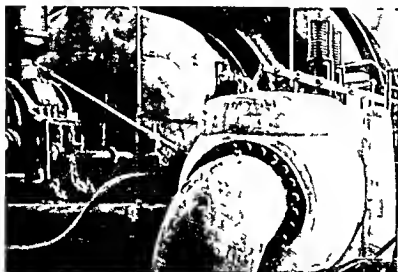
Fig 113—Asbestos blankets used for the removable port ions of turbine insulation (*Courtesy of the Johns-Manville Co*)

The most commonly used fabrics are galvanized wire netting expanded metal lath and rib lath Blankets with a metal facing on one side only are used on surfaces having irregularities such as rivet heads bolts or laps Galvanized wire netting is used as a facing where considerable flexibility is required for bending or fitting to irregular shapes Expanded metal lath is somewhat more rigid than wire netting and offers an excellent base for insulating or finishing cement Rib lath is used where maximum rigidity is required and is available with the ribs turned either in or out Turning the ribs out provides an air space between the insulation and the hot surface an advantage utilized in insulating steel breeching shells The same type of facing is

not always used on both sides of the blankets, various combinations of facings being available. These blankets are used for temperatures up to 1000°F. and are made in standard sizes of 2 by 4 ft. and 2 by 8 ft., in thicknesses from 1 to 6 in.

Insulating blankets are also available with muslin, canvas, asbestos-paper, or kraft-paper facings. Muslin or canvas facings are used on blankets for insulating busses and railroad cars. Kraft-paper facings are used on blankets for insulating equipment exposed to excessive moisture, such as refrigerator cars. These types of blankets are available in any length, in widths up to 10 ft., and in thicknesses up to 4 in.

Another type of blanket used where readily removable insulation is desired consists of a quilted asbestos-cloth envelope filled with loose asbestos fiber and tufted with copper wire. These blankets are used for temperatures up to 900°F. on irregular surfaces, such as steam-turbine casings, paper-digester shells, or valve bonnets. Figure 113 illustrates the use of such blankets for the removable portions of the insulation on a steam turbine.

57. Fillers.—Insulating fillers are of two types: those in powder or granular form which can be poured and those in the form of loose fibers which must be packed into place by hand. Either of these types is adapted for use where structural strength and rigidity are not required of the insulation or where lack of space prohibits the use of any solid form of insulation.

Natural *diatomaceous silica* in powdered form can be poured readily and will withstand temperatures up to 1600°F. It is particularly suitable for use between refractory linings and steel, concrete, or brick shells. It is also used for covering the tops of heated equipment such as ovens, furnaces, or kilns. In such applications it is usually tamped lightly into place to prevent subsequent settling, the tamping being facilitated by moistening the powder. To prevent dusting, a lime or cement slurry is used to form a top crust. Powdered diatomaceous silica is also calcined for use at temperatures up to 2000°F.

Loose *rock wool* or *glass wool* may be used where the insulation can be packed into place. A high density of packing to prevent settling from vibration is not necessary because of the natural resilience of these materials. Rock and glass wool are also made in granulated form for use where the insulation must be poured into place. Both the loose and the granulated forms

have an advantage over powders in that they are free from dust ing and do not sift through cracks They can be used for temperatures up to 1000°F

Other fillers that are used for high-temperature work are loose asbestos fiber, mixtures of powdered diatomaceous silica and asbestos fiber, and expanded vermiculite For low temperature applications, ground cork, loose rock wool impregnated with asphalt, and granulated rock wool treated with a bituminous binder are used

58 Cements—Insulating cements are generally used for filling cracks and finishing surfaces that are covered with molded insulation or for covering small irregular surfaces Certain types can be used, however, for the construction of a monolithic insulation when the shape or inaccessibility of the equipment makes the use of sheets or blocks impractical The various cements all consist of some insulating material used as a base and mixed with a suitable binder These cements are furnished dry and must be mixed with water

85% Magnesia cement is the same material used in making molded 85% Magnesia insulation, except that it is in powdered form It is a better insulator than other cements but is not ordinarily recommended either as a finishing cement or for insulating large areas It can be used for temperatures up to 600°F

Asbestos cements consist of asbestos fiber mixed with a binding material Several grades are available Those which produce a smooth, hard surface are used principally as a finish over sheet or block insulation Those which are more porous are used for sealing joints and for insulating irregular surfaces, such as valves, flanges, and other pipe fittings These cements can be used for temperatures up to 1000°F

Rock or *glass wool* forms the base for a cement that is frequently used in place of sheets or blocks to construct a monolithic insulation, as shown in Fig 114 If the temperature does not exceed 1200°F , this insulation can be removed, remixed with water, and reapplied If reapplication is not essential, it can be used up to 1500°F A reasonably smooth finish can be obtained by troweling

Expanded vermiculite is the base for another cement that can be used to construct monolithic insulations This cement is

not quite so good an insulator as the rock- or glass-wool cements, but it can be used at higher temperatures. It can be taken off, remixed, and reapplied if the temperature does not exceed 1500°F. If this is not essential, it can be used up to 1800°F.

Powdered *diatomaceous silica* mixed with asbestos fiber is the base for a cement used to point up and fill cracks on the corresponding type of molded insulation. This cement is not
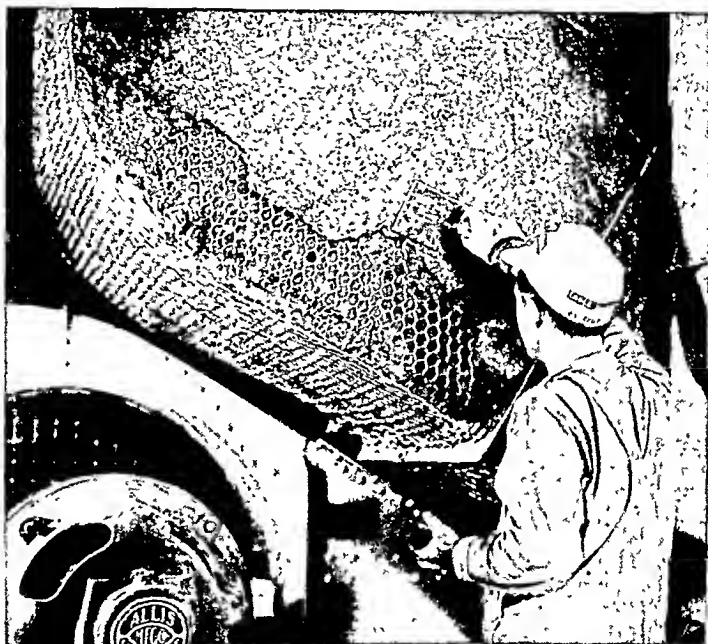
Fig. 114.—Showing the application of rock-wool insulating cement to form a monolithic covering. (*Courtesy of the Eagle-Picher Co.*)

generally used for hard finishing or for insulating large areas. It can be used up to 1600°F., or, if the diatomaceous silica has been calcined, up to 1900°F.

Weatherproof cements made of asbestos fibers and asphalt are used for protecting the insulation on outdoor equipment. These cements are supplied in plastic form and can be used at temperatures up to 400°F. They are usually applied in a single coat $\frac{1}{4}$ in. thick over a coat of insulating cement.

Adhesive cements for temporarily holding sheet, block or brick insulation in place until the outer binding support is applied are also made. Where the temperatures do not exceed 500°F and where wire reinforcement is impractical, these cements may be used for permanently bonding the blocks to the metal surface or to each other.

59. Insulating Concrete—Coarsely ground calcined diatomaceous silica is used both as an insulating filler, as described in Sec 57, and as an ingredient in making insulating concrete. Four parts of the diatomaceous silica are mixed with one part of Portland cement by volume and enough water is added to form a plastic mass. The resulting concrete can be cast into any desired shape or it can be applied to steel surfaces by means of a cement gun. It has a thermal conductivity less than one-third that of fire-clay brick and can be exposed directly to temperatures as high as 1800°F. It can be used for the construction of enameling ovens furnace doors dampers baffles and other heated equipment subjected to comparatively low temperatures.

60. Insulating Brick—Insulating bricks can be used in the construction of high temperature equipment such as heat treating furnaces boiler settings soaking pits and kilns. Two types are available one used purely as an insulator and requiring the protection of a firebrick facing and the other used as a combination insulator and refractory. The latter type can be used to replace firebrick in furnaces where the walls are not subjected to slag action or mechanical abrasion. This construction permits the use of thinner walls with lower heat storage capacities and thereby decreases both the time required for heating up and the heat lost in cooling down the furnaces. A typical installation using the first type of brick is shown in Fig 4 in Chap I.

Various materials are used to make insulating bricks. One kind of brick is cut from natural diatomaceous silica. Another kind is made from ground diatomaceous silica pressed into bricks and calcined. Other kinds consist of refractory clay as a base, with various fibrous or cellular materials added to reduce the *thermal conductivity*. Insulating bricks are available in all standard firebrick sizes and shapes.

61. Method of Calculating Heat Loss through Insulation—The rate of heat loss through insulation on *flat surfaces* can be calculated by the equation

$$q = \frac{kA(t_s - t_a)}{L + (k/h_t)}, \tag{59}$$

where q = the rate of heat loss, B.t.u. per hr.

k = the thermal conductivity of the insulation at its average temperature (approximately the average of t_s and t_a), B.t.u./(ft.)(hr.)(°F.).

A = the area of the surface that is covered with insulation, sq. ft.

t_s = the temperature of the surface that is covered with insulation, °F.

t_a = the temperature of the air surrounding the insulation, °F.

L = the thickness of the insulation, ft.

h_t = the "combined coefficient" for convection and radiation from the surface of the insulation (see Table 59), B.t.u./(sq. ft.)(hr.)(°F.).

Although the combined coefficient h_t depends upon the temperature of the surface of the insulation, sufficiently accurate values for practical purposes can usually be obtained if h_t is evaluated at any reasonable, assumed surface temperature. If more accurate values are required, the exact surface temperature can be calculated by a trial-and-error solution, as shown in the following problem:

Illustrative Problem 1.—A flat, vertical surface at a temperature of 500°F. is covered with 2 in. of 85% Magnesia. Calculate the exact rate of heat loss per square foot of surface if the surrounding air temperature is 70°F.

Solution.—As a first approximation, assume the surface temperature of the insulation to be 100°F. Based on this assumption, the average temperature of the insulation is 300°F.; the thermal conductivity of 85% Magnesia at this average temperature is 0.043 B.t.u./(ft.)(hr.)(°F.); and the combined coefficient h_t, from Table 59, is 1.68 B.t.u./(sq. ft.)(hr.)(°F.). Hence, by Eq. (59),

$$q = \frac{0.043 \times 1 \times (500 - 70)}{(\frac{2}{12}) + (0.043/1.68)}$$
$$= 96.1 \text{ B.t.u./(sq. ft.)(hr.).}$$

If greater accuracy is desired, a more accurate value of the temperature t_s' of the surface of the insulation can be calculated by the equation

$$q = h_t A(t_s' - t_a).$$

Thus,

$$96.1 = 1.68 \times 1 \times (t_s' - 70),$$

or

$$t_s' = 127°F$$

If the calculations are repeated using this surface temperature, $q = 97.5$ Btu/(sq ft)(hr) is obtained

TABLE 59—COMBINED COEFFICIENT h_c FOR CONVECTION AND RADIATION FROM FLAT OR CYLINDRICAL SURFACES IN A ROOM AT 70°F

The following table is calculated by the equation

$$h_c = h + eh_r$$

where h_c = the "combined coefficient" for convection and radiation, Btu/(sq ft)(hr)(°F)

h = the film coefficient of convection (obtained from Case 12-14 or 18 in Chap IV) Btu/(sq ft)(hr)(°F)

e = the emissivity of the surface of the insulation (assumed = 0.90)

h_r = the radiation coefficient (calculated by Eq (21) in Chap III), Btu/(sq ft)(hr)(°F)

Type of surface	Temperature of surface				
	100°F	150°F	200°F	250°F	300°F
	h_c Btu (sq ft)(hr)(°F)				
Flat surfaces					
Vertical	1 68	2 07	2 38	2 67	2 95
Horizontal facing upward	1 86	2 32	2 62	2 99	3 28
Horizontal facing downward	1 46	1 77	2 03	2 29	2 54
Cylindrical surfaces					
Vertical	1 68	2 07	2 38	2 67	2 95
Horizontal 2 in o d	1 98	2 40	2 73	3 03	3 32
Horizontal 4 in o d	1 82	2 20	2 51	2 79	3 06
Horizontal 6 in o d	1 75	2 10	2 40	2 66	2 93
Horizontal 8 in o d	1 69	2 03	2 32	2 58	2 85
Horizontal 10 in o d	1 66	1 99	2 27	2 53	2 79
Horizontal 12 in o d	1 63	1 95	2 23	2 48	2 74

The rate of heat loss through insulation on *pipes* or other cylindrical surfaces can be calculated by the equation

$$q = \frac{12\pi l (t_i - t_s)}{2.3 \log_{10}(r_2/r_1) + (k_i/h_c r_2)}$$ (60)

where r_1 = the inside radius of the insulation ft

r_2 = the outside radius of the insulation ft,

l = the length of the pipe, ft

and the other symbols have the same meanings as for Eq (59)

An approximate value for the rate of heat loss through insulation can be calculated by Eq. (3) or (4) in Chap. I if the temperature of the outside surface of the insulation is assumed to be the same as the temperature of the surrounding air. This procedure may introduce a considerable error if the thickness of the insulation is small or the temperature difference $(t_s - t_a)$ is large.

Illustrative Problem 2.—A horizontal 3-in. nominal-diameter pipe is covered with 1 in. of corrugated asbestos insulation (four ply per in.). If the temperature of the pipe is 300°F. and that of the surrounding air is 70°F., calculate the heat loss from the pipe per foot of length.

Solution.—The actual outside diameter of the pipe is 3.500 in.; so the outside diameter of the insulation is 5.5 in. Assuming that the surface temperature of the insulation is 125°F., the average temperature of the insulation is 213°F.; the thermal conductivity is 0.060 B.t.u./(ft.)(hr.)(°F.); and the combined coefficient h_t, from Table 59, is 1.95 B.t.u./(sq. ft.)(hr.) (°F.). Hence, by Eq. (60),

$$q = \frac{0.060 \times 2 \times 3.14 \times 1 \times (300 - 70)}{2.3 \log_{10} (2.75/1.75) + (0.060 \times 12)/(1.95 \times 2.75)}$$
$$= 148 \text{ B.t.u./(hr.)(ft. of length).}$$

Approximate Solution.—If the surface temperature of the insulation is assumed to be the same as the temperature of the surrounding air, the heat loss can be calculated by Eq. (4) in Chap. I:

$$q = \frac{0.060 \times 2 \times 3.14 \times 1 \times (300 - 70)}{2.3 \log_{10} (2.75/1.75)}$$
$$= 192 \text{ B.t.u./(hr.)(ft. of length).}$$

This value is approximately 30 per cent too high.

Equation (59) can be derived as follows: Letting t_s' denote the temperature of the outside surface of the insulation, the rate of heat transfer through the insulation by conduction [Eq. (3) in Chap. I] is

$$q = \frac{kA(t_s - t_s')}{L}.$$

Assuming steady flow conditions, this rate is equal to the rate at which heat is lost from the surface of the insulation by convection [Eq. (23) in Chap. III] and by radiation [Eq. (10) in Chap. II]:

$$q = hA(t_s' - t_a) + h_r A(t_s' - t_a)e.$$

or

$$q = h_t A(t_s' - t_a).$$

Equation (59) is obtained by eliminating t_s' from these equations
Equation (60) can be derived from Eqs (4), (10), and (23) in a
similar manner

62. Most Economical Thickness of Insulation.—As the thickness of insulation on any surface is increased, the rate of heat loss from the surface is decreased, but the cost of the insulation is increased   The most economical thickness of insulation is
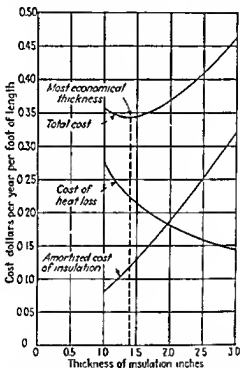
FIG 115 —Determination of the most economical thickness of insulation
(Data given in Illustrative Problem 3)

that for which the sum of the yearly cost of the heat loss plus
the yearly cost of the insulation is a minimum   The method
of calculation is illustrated in the following problem

Illustrative Problem 3 — A 3-in nominal diameter steam pipe is to be
covered with 85% Magnesia   Determine the most economical thickness of
insulation under the following conditions

Temperature of the pipe = 400°F
Temperature of the surrounding air = 80°F

Time of operation per year	$= 8$ months
	$= 5,760$ hr.
Cost of the heat lost	$= \$0.30$ per 10^6 B.t.u.
Cost of the insulation (applied):	
1-in. thickness	$= \$0.53$ per ft.
$1\frac{1}{2}$-in. thickness	$= \$0.86$ per ft.
2-in. thickness	$= \$1.25$ per ft.
3-in. thickness	$= \$2.13$ per ft.

Fraction of the cost of the insulation to be amortized each year $= 0.15$.

Solution.—The heat loss per hour per foot of length can be calculated for each thickness of insulation by Eq. (60). The yearly cost of the heat loss is equal to the product of this heat loss per hour times the hours of operation per year times the cost of the heat per B.t.u. The yearly cost of the insulation is equal to the product of the cost of the insulation (applied) times the fraction of this cost to be amortized each year. The results obtained from these calculations are shown in Fig. 115. For the data given, it would be most economical to cover the pipe with $1\frac{1}{2}$ in. of insulation.

If the cost of the insulation per cubic foot is approximately constant for the different thicknesses, the most economical thickness can be calculated by the following equations developed by Patton:[1]

For *flat* surfaces,

$$L_0 = 12 \left[\sqrt{\frac{nc'k(t_s - t_a)}{fc \cdot 10^6}} - \frac{k}{h_t} \right], \tag{61}$$

where $L_0 =$ the most economical thickness of insulation, in.

       $n =$ the time that the equipment operates, hr. per year.

       $c =$ the cost of the insulation (applied), dollars per cu. ft.

       $c' =$ the cost of the heat lost, dollars per million B.t.u.

       $f =$ the fraction of the cost of the insulation to be amortized each year.

       $k =$ the thermal conductivity of the insulation, B.t.u./(ft.)(hr.)(°F.).

       $h_t =$ the "combined coefficient" for convection and radiation from the surface of the insulation (see Table 59), B.t.u./(sq. ft.)(hr.)(°F.)

       $t_s =$ the temperature of the surface to be insulated, °F.

       $t_a =$ the temperature of the air surrounding the insulation, °F.

For *cylindrical* surfaces, the most economical ratio of outside to inside radius of insulation $(r_2/r_1)_0$ can be determined from Table

[1] *Heating, Piping, and Air Conditioning,* vol. 4, p. 6, 1932.

60 after the quantity K_1 has been calculated by the equation

$$K_1 = \frac{1}{r_1}\left[\sqrt{\frac{nc'\lambda(t_s - t_a)}{fc} \times 10^6} - \frac{k}{h_1}\right],$$ (62)

where r_1 = the inside radius of the insulation, in

r_2 = the outside radius of the insulation, in

and the other symbols have the same meanings as for Eq (61)

TABLE 60—MOST ECONOMICAL RATIO OF OUTSIDE TO INSIDE RADIUS OF INSULATION

K_1 calculated by Eq (62)

K_1	$\left(\frac{r_2}{r_1}\right)_e$	K_1	$\left(\frac{r_2}{r_1}\right)_e$
0 00	1 000	0 24	2 799
0 02	1 218	0 26	2 916
0 04	1 407	0 28	3 031
0 06	1 578	0 30	3 144
0 08	1 737		
0 10	1 888	0 40	3 68
0 12	2 032	0 50	4 19
0 14	2 170	0 60	4 67
0 16	2 303	0 70	5 13
0 18	2 431	0 80	5 58
0 20	2 556	0 90	6 02
0 22	2 679	1 00	6 45

Illustrative Problem 4—Using the data given in Problem 3 determine the most economical thickness of insulation by means of Eq (62) and Table 60

Solution —The actual outside diameter of a 3-in pipe is 3 500 in , so the inside radius of the insulation is 1 75 in The average cost per cubic foot of the four thicknesses of insulation given in Problem 3 is approximately \$5 20 Assuming that the surface temperature of the insulation will be about 125°F the average thermal conductivity of the insulation will be 0 042 B t u /(ft)(hr)(°F) and the combined coefficient h from Table 58, will be approximately 1 90 B t u (q ft)(hr)(°F) Hence by Eq (62),

$$K_1 = \frac{1}{1\ 75}\left[\sqrt{\frac{5\ 760 \times 0\ 30 \times 0\ 042 \times (400 - 70)}{0\ 15 \times 5\ 20 \times 10^6}} - \frac{0\ 042}{1\ 90}\right]$$

$$= 0\ 0875$$

Therefore, from Table 60, the most economical ratio of outside to inside radius of insulation is

$$\left(\frac{r_2}{r_1}\right)_e = 1\ 794,$$

and

$$r_2 = 1.75 \times 1.794$$
$$= 3.14 \text{ in.}$$

Finally, the most economical thickness of insulation is $3.14 - 1.75 = 1.39$ in.

Patton showed that, approximately,

$$K_1 = \frac{1}{12}\left(\frac{r_2}{r_1}\right)_0 \log_e \left(\frac{r_2}{r_1}\right)_0.$$

Table 60 was calculated from this equation.

63. Thickness of Insulation Required to Prevent Freezing.— The thickness of insulation required to prevent water freezing in pipes can be calculated by the equation

$$r_2 = r_1 \text{ antilog}_{10}\left[\frac{2.73kl(t_w - t_a)}{G(t_w - 32)}\right], \tag{63}$$

where r_1 = the inside radius of the insulation, in.

   r_2 = the outside radius of the insulation, in.

   k = the thermal conductivity of the insulation, B.t.u./ (ft.)(hr.)(°F.).

   l = the length of the pipe, ft.

   t_w = the temperature of the water entering the pipe, °F.

   t_a = the temperature of the air surrounding the pipe, °F.

   G = the rate at which the water flows, lb. per hr.

Illustrative Problem 5.—A $1\frac{1}{2}$ in. water pipe is to be protected from freezing by a covering of hair felt. The water enters the pipe at 50°F., and the surrounding air is at -10°F. Calculate the thickness of insulation required if the water flows at a minimum rate of 5 gal. per hr.

*Solution.—*The actual outside diameter of a $1\frac{1}{2}$-in. pipe is 1.900 in.; so the inside radius of the insulation is 0.95 in. The thermal conductivity of hair felt is 0.022 B.t.u./(ft.)(hr.)(°F.), and the weight rate of flow of the water is $5 \times 8.35 = 41.8$ lb. per hr. By Eq. (63),

$$r_2 = 0.95 \text{ antilog}_{10}\left[\frac{2.73 \times 0.022 \times 100 \times (50 + 10)}{41.8 \times (50 - 32)}\right]$$
$$= 2.86 \text{ in.}$$

Hence, the insulation must be at least $2.86 - 0.95 = 1.91$ in. thick.

If there is no flow of water through the pipe and the surrounding air temperature t_a remains below 32°F. indefinitely, freezing cannot be prevented regardless of the thickness of insulation used. The time required for water initially at

temperature t_w to be cooled to 32°F when there is no circulation can be calculated by the equation

$$\theta = \left(\frac{t_w - 32}{t_w - t_a}\right)\left(\frac{23}{l'} + 0.04w\right)\frac{\log_{10}(r_2/r_1)}{k},\qquad (64)$$

where θ = the time required for the water to cool to 32°F., hr

l' = the length of pipe containing 1 cu ft of water (see Table VI in the Appendix), ft

w = the weight of the pipe (see Table VI in the Appendix), lb per ft of length

t_w = the initial temperature of the water, °F.,

and the other symbols have the same meanings as for Eq. (63)

Illustrative Problem 6—A 1½-in wrought iron water pipe is covered with 2 in of hair felt If the surrounding air temperature is −10°F, calculate the time required for water standing in the pipe to cool from an initial temperature of 50°F to a temperature of 32°F

Solution—The actual outside diameter of 1½-in pipe is 1.900 in , so the inside radius of the insulation is 0.95 in and the outside radius is 2.95 in One foot of the pipe weighs 2.717 lb and 70.7 ft contains 1 cu ft of water The thermal conductivity of hair felt is 0.022 Btu/((t)(hr)(°F) Hence the time θ required for the water to cool to the freezing point is by Eq (64)

$$\theta = \left(\frac{50 - 32}{50 + 10}\right)\left(\frac{23}{70.7} + 0.04 \times 2.717\right)\frac{\log_{10}(2.95/0.95)}{0.022}$$

$$= 2.91 \text{ hr}$$

Equation (64) can be derived by equating the rate at which the water gives up heat in cooling from t_w to 32°F to the rate at which heat is transferred by conduction through the insulation In determining the latter the surface temperature of the insulation is assumed to be equal to t_a and the change in temperature of the water is neglected Each of these assumptions results in an increased, and therefore more conservative, value for r_2

Equation (64) can be derived by dividing the amount of heat given up by the water and by the metal pipe in cooling from t_w to 32°F by the rate at which heat is transferred by conduction through the insulation *the same assumptions being made as for Eq (63)*

64 Thickness of Insulation Required to Prevent Sweating— The ratio of outside to inside radius of insulation (r_2/r_1) required to prevent sweating on pipes can be determined from Table 61 after the quantity K_x has been calculated by the equation

$$K_2 = \frac{8k}{r_1}\left(\frac{t_d - t_w}{t_a - t_d}\right), \tag{65}$$

where r_1 = the inside radius of the insulation, in.

r_2 = the outside radius of the insulation, in.

k = the thermal conductivity of the insulation, B.t.u /
(ft.)(hr)(°F).

t_a = the dry-bulb temperature of the air surrounding the
insulation, °F.

t_d = the dew-point temperature of the air surrounding the
insulation, °F.

t_w = the temperature of the fluid inside the pipe, °F.

TABLE 61 —RATIO OF OUTSIDE TO INSIDE RADIUS OF INSULATION REQUIRED
TO PREVENT SWEATING ON PIPES

K_2 calculated by Eq. (65)

K_2	$\left(\frac{r_2}{r_1}\right)$	K_2	$\left(\frac{r_2}{r_1}\right)$
0 00	1 000	1 20	2 741
0 10	1 210	1 30	2 854
0 20	1 393	1 40	2 965
0 30	1 559	1 50	3.075
0 40	1 713		
0 50	1 858	2 00	3 60
0 60	1 997	2 50	4 09
0 70	2 130	3 00	4 56
0 80	2 259	3 50	5 01
0 90	2 384	4 00	5 44
1 00	2 506	4 50	5 86
1 10	2 625	5 00	6 27

Illustrative Problem 7.—A 2-in. pipe carrying water at 50°F is to be
covered with sufficient wool felt to prevent sweating   Calculate the thick-
ness of insulation required if the temperature of the surrounding air is 90°F.
and the relative humidity is 70 per cent.

Solution.—The actual outside diameter of a 2-in pipe is 2 375 in.; so the
inside radius of the insulation will be 1.188 in   The dew-point tempera-
ture corresponding to a dry-bulb temperature of 90°F. and a relative
humidity of 70 per cent is 79°F. (determined from any psychrometric chart).
The average temperature of the insulation will be 65°F, and its thermal
conductivity at this temperature will be 0 024 B t u /(ft.)(hr.)(°F.).   By
Eq. (65),

$$K_2 = \frac{8 \times 0\ 024}{1.188}\left(\frac{79 - 50}{90 - 79}\right)$$
$$= 0\ 426.$$

Therefore from Table 61, the ratio of the outside to the inside radius of the insulation must be

$$\frac{r_2}{r_1} = 1\,751,$$

or

$$r_2 = 1\,751 \times 1\,188$$
$$= 2\,08\text{ in}$$

Hence, the thickness of the insulation must be at least

$$2\,08 - 1\,188 = 0\,89\text{ in}$$

Equation (65) can be derived as follows The minimum thickness of insulation required to prevent sweating is that thickness for which the surface temperature is exactly equal to the dew-point temperature of the surrounding air This thickness can be found by equating the rate of heat transfer through the insulation by conduction to the rate of heat transfer from the surface of the insulation to the surrounding air by convection and by radiation Thus, per foot of pipe,

$$\frac{k2\pi(t_i - t_o)}{2\,3\log_{10}(r_2/r_1)} = h_i\left(\frac{2\pi r_2}{12}\right)(t_o - t_d),$$

where h_i is the "combined coefficient" for convection and radiation (see Table 59) If h_i is assumed equal to 0 65 B t u / (sq ft)(hr)(°F) [the smaller this value the more conservative the result] this equation reduces to

$$\left(\frac{r_2}{r_1}\right)\log_{10}\left(\frac{r_2}{r_1}\right) = \frac{8k}{r_1}\left(\frac{t_d - t_o}{t_o - t_d}\right)$$

Equation (65) is obtained by letting $K_2 = (r_2/r_1)\log_{10}(r_2/r_1)$
Table 61 is based on the latter equation defining K_2

APPENDIX

TABLE I.—THERMAL CONDUCTIVITIES OF METALS*
Linear interpolation between temperatures is permissible

Metal	Temperature, °F	Thermal conductivity k B t u /(ft)(hr)(°F)
Aluminum	0	116
	200	119
	400	124
	600	134
	900	153
Antimony	32	10 6
	212	9 7
Bismuth	64	4 7
	212	3 9
Brass (70 30)	0	55
	800	68
Bronze	68	33 6
	210	41 0
Cadmium	64	53 7
	212	52 2
Cast iron	32	29
	212	28
Constantan	64	13 1
	212	15 5
Copper (pure)	0	225
	200	218
	1200	202
Gold	64	169
	212	170
Iron (pure)	64	39 0
	212	36 6
Lead	0	21
	400	19
Magnesium	32	92
	212	92
Mercury	32	3 6
	122	4 6
Nickel	0	35
	600	32
Nickel silver	32	16 9
	212	21 5
Platinum	64	40 2
	212	41 9
Silver	32	242
	212	238
Steel (mild) cold rolled	200	26
	1100	21
Tin	32	37
	392	34
Tungsten	68	92 5
Wrought iron	64	34 9
	212	34 6
Zinc	0	65
	400	61
	800	54

* Compiled from "International Critical Tables," Marks "Mechanical Engineers Handbook," and other sources

TABLE II—THERMAL CONDUCTIVITIES OF INSULATING MATERIALS[a]
Linear interpolation between temperatures is permissable   Section numbers
refer to the section in which the material is described

Material	Apparent density ρ lb per cu ft	Temperature °F	Thermal conductivity k B.t.u./(ft)(hr)(°F)
Pipe covering and blocks			
85% Magnesia (Secs 50b and 53a)	17	100	0 039
	17	500	0 047
Diatomaceous silica, asbestos and bonding material (Secs 50c and 53a)	23	100	0 045
	23	1000	0 083
Laminated asbestos felt (40 1 ly per in) (Secs. 50d and 53a)	30	100	0 033
	30	500	0 048
Laminated asbestos felt (20 ply per in)		100	0 045
		500	0 065
Felted asbestos (Secs 50c and 53a)	36	100	0 097
	36	200	0 110
	36	300	0 118
	36	800	0 150
	44	−300	0 100
	44	− 50	0 132
	44	32	0 135
Felted rock or glass wool (Secs 50f and 60p)	16	100	0 030
	16	600	0 067
Corrugated asbestos (4 ply per in) (See 50f)	12	100	0 050
	12	300	0 068
Wool felt (Secs. 50k and 52r)	8 5	32	0 072
	8 5	200	0 033
	20 6	86	0 03
Cork molded (Secs 52b and 54a)	7 5	32	0 021
	7 5	122	0 074
	12 5	32	0 030
	12 5	122	0 031
Hair felt (Sec 52d)	25	32	0 025
	15	60	0 077
	9	70	0 022
	17	70	0 022
Crumpled aluminum foil (⅜₂ in air spaces) (Sec 50f)	0 2	100	0 025
	0 2	350	0 038
Sheets and blankets			
Asbestos millboard (See 55)	55	124	0 096
Asbestos sheets (Sec 55)	36	100	0 097
	36	200	0 110
	36	300	0 118
	36	500	0 130
	44	32	0 135
Corrugated asbestos (4 ply per in.) (Sec 55)	12	100	0 050
	12	300	0 068
Hair felt (Sec 55)	9	70	0 022
	17	70	0 022
Hair and asbestos fiber felted	7 8	70	0 023

* Compiled from International Critical Tables," Marks Mechanical Engineers Hand book and other sources

TABLE II—THERMAL CONDUCTIVITIES OF INSULATING MATERIALS *—(Continued)

Material	Apparent density ρ lb per cu ft	Temperature, °F	Thermal conductivity k, B t u /(ft) (hr)(°F)
Sheets and blankets (Continued)			
Wool felt	8 5	32	0 022
	8 5	200	0 023
	20 6	86	0 03
Aluminum foil ($\frac{3}{8}$-in spacing) (Sec 55)	0 2	100	0 025
	0 2	350	0 038
Felted rock or glass wool (Sec 56)	16	100	0 030
	16	600	0 057
Fillers			
Diatomaceous silica, powdered (Sec 57)	10 6	70	0 026
	17 2	100	0 032
	17 2	1600	0 074
	20 0	100	0 036
	20 0	1600	0 082
Rock or glass wool loose or granular (Sec 57)	6	70	0 022
	18	70	0 024
Asbestos, loose (Sec 57)	29 3	−300	0 054
	29 3	32	0 089
	29 3	210	0 093
Cork, ground (Sec 57)	10 0	32	0 021
	10 0	200	0 032
Cork, regranulated (Sec 57)	8 1	86	0 026
Magnesia powdered	49 7	117	0 35
Gypsum, powdered	26	70	0 047
	34	70	0 047
Wool (animal), loose	6 9	86	0 021
Wood fiber, chemically treated	2 2	70	0 023
Insulating cements			
85 % Magnesia (Sec 58)			0 6
Asbestos (Sec 58)			1 2
Asbestos and rock wool (Sec 58)			0 9
Insulating concrete			
Four parts diatomaceous silica and one part cement (Sec 59).	61 8	400	0 16
	61 8	1600	0 23
Insulating brick			
Diatomaceous silica, natural perpendicular to strata (Sec 60)	27 7	400	0 051
	27 7	1600	0 077
Diatomaceous silica, natural, parallel to strata (Sec 60)	27 7	400	0 081
	27 7	1600	0 106
Diatomaceous silica, calcined (Sec 60)	38	400	0 14
	38	1600	0 18
Diatomaceous silica and fire clay, calcined (Sec 60)	42 3	400	0 14
	42 3	1600	0 19
Kaolin insulating brick	27	932	0 15
	27	2100	0 26

* Compiled from "International Critical Tables " Marks "Mechanical Engineers' Hand book," and other sources

TABLE IV.—EMISSIVITY OF SURFACES *—(Continued)

Surface	Temperature,† °F	Emissivity†
Nickel alloy *(Continued)*		
Same after 24 hr at 980°F	420– 980	0 62–0 73
NCT-3 (20%i, 25Cr, brown aſ‌lotched, oxidised from service)	420– 980	0 90–0 97
NCT-6 (60%Ni 12Cr smooth black firm adhesive oxide coat from service)	520–1045	0 89–0 82
Platinum		
Polished plate	440–2560	0 054–0 17
Platinum filament	80–2240	0 036–0 192
Silver, pure polished	440–1160	0 02–0 032
Tantalum filament	2420–5430	0 194–0 31
Tin bright	76	0 047–0 064
Tungsten aged filament	80–6000	{ 0 032–0 35 ; 0 018
Zinc		
Commercial polish	440– 620	0 045–0 053
Oxidised at 750°F	750	0 11
Galvanised iron fairly bright	82	0 23
Galvanised iron gray oxidised	75	0 28

Refractories Building Materials Paints Miscellaneous

Aluminium paints (vary with amount of lacquer body and age)	212	0 27–0 67
Asbestos	100– 700	0 93–0 95
Carbon		
Candle soot lampblack—waterglass	70– 700	0 95± 01
Oil		
Lubricating layer 0 01 in thick	68	0 82
Linseed one and two coats on Al foil	68	0 56 and 0 57
Rubber soft gray reclaime ‡	76	0 859
Miscellaneous I shiny black lacquer planed oak white enamel serpentine gypsum white enamel paint roofing paper lime plaster black matte shellac	70	0 87–0 91
Miscellaneous II glazed porcelain white paper fused quartz polished marble rough red brick smooth glass hard glossy rubber flat black lacquer water electrographite	70	0 92–0 96

* Prepared by H C Hottel

† When two temperatures and two emissivities are given they correspond first to first and second to second and linear interpolation is permissible

TABLE V.—DIMENSIONS OF FREQUENTLY USED SIZES OF TUBING

Outside tube, diameter, in	Wall thickness		Inside tube diameter in	Surface per ft of length		Length of tube per sq ft of surface		Cross sectional area	
	Birmingham Wire Gauge (B W G)	In		Outside surface sq ft per ft of length	Inside surface, sq ft per ft of length	Outside, ft	Inside ft	Outside, sq in	Inside, sq in
½	20	0 035	0 430	0 13090	0 11257	7 6394	8 8831	0 19635	0 14522
	19	0 042	0 416		0 10891		9 1820		0 13592
⅝	18	0 049	0 527	0 16362	0 13797	6 1116	7 2480	0 30680	0 21813
	16	0 065	0 495		0 12959		7 7166		0.19244
	14	0 083	0 459		0 12017		8 3218		0 16547
¾	18	0 049	0 652	0 19635	0 17069	5 0930	5 8585	0 44179	0 33388
	16	0 065	0 620		0 16232		6 1608		0 30191
	14	0 083	0 584		0 15289		6 5406		0 26787
⅞	18	0 049	0 777	0 22907	0 20342	4 3654	4 9160	0 60132	0 47417
	16	0 065	0 745		0 19504		5 1271		0 43592
	14	0 083	0 709		0 18562		5 3875		0 30481
1	18	0 049	0 902	0 26180	0 23614	3 8197	4 2347	0 78540	0 63900
	16	0 065	0 870		0 22777		4 3905		0 59447
	14	0 083	0 834		0 21834		4 5800		0 54629
1¼	18	0 049	1 152	0 32725	0 30159	3 0558	3 3157	1 2272	1 04231
	16	0 065	1 120		0 29322		3 4105		0 98520
	14	0 083	1 084		0 28379		3 5237		0 92289
1½	16	0 065	1 370	0 39270	0 35867	2 5465	2 7881	1 7672	1 4741
	14	0 083	1 334		0 34924		2 8634		1 3977
2	14	0 083	1 834	0 52360	0 48014	1 9099	2 0827	3 1416	2 6417
	12	0 109	1 782		0 46653		2 1435		2 4941

TABLE VI.—Standard Dimensions for Standard-weight Wrought-iron Pipe*

Nominal size, in.	Nominal size, mm	Actual diameters		Nominal thick- ness, in.	Circumference		Transverse areas			Length of pipe per sq. ft.		Length of pipe con- taining cu. ft.	Nominal weight, lb.		Number of threads per inch of screw
		External, in.	Approx- imate internal, in.		External, in.	Internal, in.	External sq. in.	Internal sq. in.	Metal sq. in.	External surface, ft.	Internal surface, ft.		Plain ends	Threaded and coupled	

*From W. H. McAdams "Heat Transmission" McGraw-Hill Book Company Inc 1933

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